The EDITOR'S PAGE

Heating · Piping and Air Conditioning

AIR conditioning is a subject that should be studied carefully by the management of the plant or building intent on improving operation and getting in shape for the keen competition which exists now and which is indicated for the future. Plenty of industrial and commercial enterprises are making money at today's reduced volume of business; they are the ones that have studied their processes and their operation and have put their equipment in shape to carry on economically and efficiently. With increased business activity, such concerns will be in a particularly advantageous position insofar as competition from not-so-completely equipped competitors is concerned.

WHAT air conditioning can do in improving a process or in putting a building in shape to earn better rentals is demonstrated emphatically by the experiences of several concerns described in a special report just issued by the general section of the *National Electric Light Association;* this report surveys opportunities for selling power for air conditioning in industrial buildings, theaters, office buildings, hotels, restaurants, stores, public buildings, etc. An investment of $61,000 for air conditioning a cotton and silk products mill saves over $14,000 annually, says this report. A cigarette manufacturer increased the output of his automatic packing, wrapping and sealing equipment by providing correct atmospheric conditions, resulting in a 400-per cent increase in production. Spending $34,000 for air-conditioning equipment, a cigar manufacturer gets a $29,000 return each year.

.....a manufacturer of effervescent salts saves $25,000 year by an investment of $36,000 in air conditioning.

.....a rayon plant's production was increased 27 per cent with air conditioning.

.....air conditioning enabled one department of an industrial chemicals plant to run throughout the year, instead of for only 200 days with consequent loss due to unproductive overhead charges.

AIR conditioning in theaters is regarded as standard and has eliminated the summer slump in this business. Restaurants find many advantages in air conditioning; patronage is increased, as is the average price of the meals served. One restaurant (see page 734 of this issue) enjoyed a 60-per cent increase in business the first summer after it installed a cooling system. A number of department stores have found air conditioning increased their sales and gave other advantages; air-conditioned office buildings have been able to get and hold tenants because of air conditioning; banks, hotels, public buildings, etc. have found air conditioning to be a paying proposition.

IN checking an existing installation of air-conditioning equipment to be sure that it is in order for efficient and economical operation, the chart and maintenance guide on page 737 of this issue will be found useful. A number of articles which have appeared from time to time in HEATING, PIPING AND AIR CONDITIONING will also be found of aid in this connection; some of them are:

"The Maintenance of Air Conditioning Equipment in Industrial Plants," by William A. Hanley, page 905, November, 1930. Vol. II, No. 11.

"Now Is the Time to Overhaul Air Cooling and Refrigerating Equipment," by R. C. Doremus, January, 1931, page 1. Vol. III, No. 1.

"Improved Compressor Operation a Road to Air Conditioning Economy," by John C. Reed and Edgar E. Ambrosius, page 746, September, 1931. Vol. III, No. 9.

"45 Per Cent Power Saving Made by Intermittent Control of Central Station Type Air-Conditioning System," by Charles S. Leopold, page 989, December, 1931. Vol. III, No. 12.

"Chart Helps Duct Changes," by George T. Condron, page 555, August, 1932. Vol. IV, No. 8.

"Economy in Drying Aided by Vapor Proofing," by Malcolm Tomlinson, page 482, July, 1932. Vol. III, No. 7.

"Turning Blades Reduce Duct Elbow Losses," by F. E. Wertheim, page 915, November, 1931. Vol. III, No. 11. Discussion, page 84, January, 1932.

"Saving Money in Motor Operation," by Louis Mackler, page 649, August, 1931. Vol. III, No. 8.

"Operation and Maintenance of Air Filters," by W. G. Frank, page 378, May, 1931. Vol. III, No. 5.

"How to Install and Care for Gages," by John A. Masek, page 727, September, 1931. Vol. III, No. 9.

MANY of the articles in this issue point the way to saving money, improving operation. The Belle City Malleable Iron Co. (page 728) uses waste-heat boilers for supplying steam used for compressing air and space heating; Herman Vetter (page 731) shows how power may be saved by proper sizing of ammonia suction lines; L. K. Ferris points out how he saves on heating water for use in the building in his charge (page 721); several articles in "On the Job" (page 742) describe practical points of help in maintenance work.

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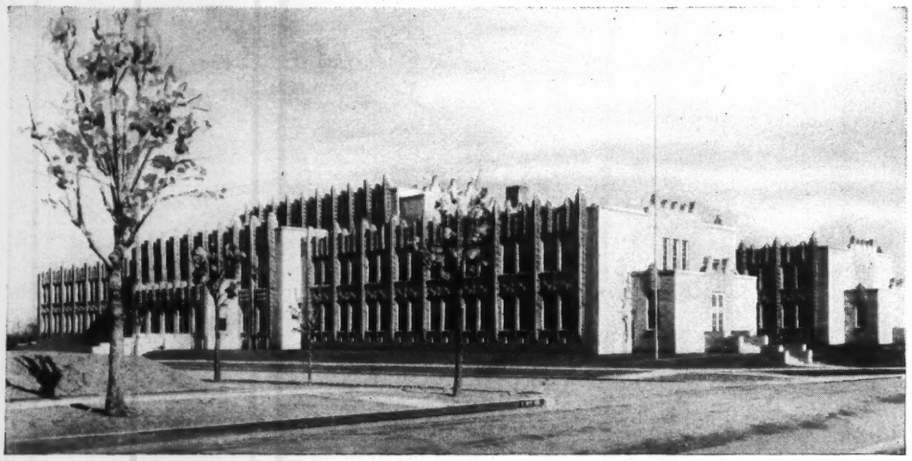

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The November, 1932

Vol. IV, No. 11

Heating · Piping and Air Conditioning

Electric Control of Motor-Driven Pumps

By Samuel R. Lewis*

NO MODERN building could be built or operated without electricity. No modern building could be built or operated without water. The modern building usually depends upon electricity to deliver the water through the agency of pumps.

The generally-used pump is continuous in speed and delivery and is rotary or centrifugal; this type lends itself admirably to electric motor propulsion. The controls discussed here are limited to those adapted to centrifugal pumps.

The engine room of a modern skyscraping office building is a fascinating place. It always is crowded with automatic electric pumps for the air-conditioning equipment and for compressed air, brine, hot water, drinking water, cold lavatory water, heating system returns, etc. The city mains rarely are under enough pressure to force water to the upper stories, and pumps are required for this service. The heating of such buildings usually is zoned, with a separate vacuum return pump for each zone.

While electricity is a wonderfully flexible and efficient servant, the control of all of these different sorts of pumps is complicated and always is important.

To indicate modern control methods the pumping equipment and its control in a modern skyscraper are described in this article.

Control for the Heating System Vacuum Pumps

The vacuum pumps for the several zones of the heating system in this building discharge the condensate into an open receiver, from which the air is released and from which the water is returned to the boilers by means of a separate centrifugal pump.

The vacuum-pump motors are constant-speed squirrel-cage induction motors of across-the-line type, since they are all of less than 5-hp. in size.

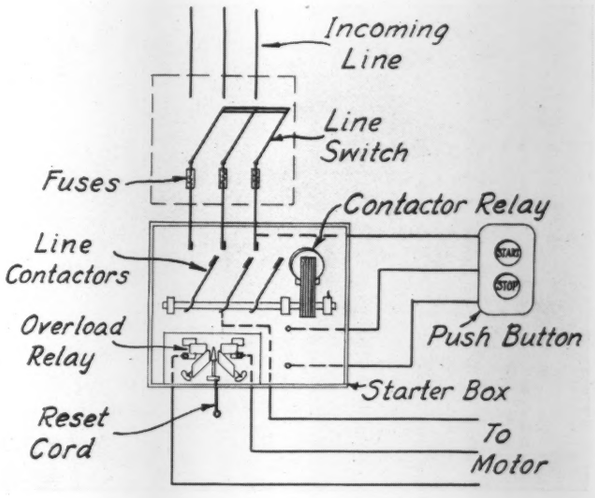
Each of the vacuum-pump motors is provided with a starter containing these features:

1. A fused line switch.
2. A magnetic contactor for closing the line to the motor and for providing under-voltage protection.
3. Temperature overload relays which will disconnect the motor from the line in case of excessive current caused by an overload on the motors.
4. A double-throw switch in one position of which the starter is controlled by the float or by the vacuum diaphragm; in the other position the starter is controlled by a push-button switch.

The above float switch and the vacuum diaphragm switches are wired in parallel.

One of these switches responds to fluctuations in the water level in the vacuum-pump receiving tank. The float switch is closed by a lever when the high level of the water in the receiving tank has been reached, and the switch is opened by the same means at the low level of the water.

The other parallel switch responds to movement of a vacuum-sensitive diaphragm on a return-pipe connection to the pump and closes at a predetermined low vacuum and opens at another predetermined higher vacuum. There is provided an adjustable tolerance limit



Wiring to a starter of the magnetic switch type for across-the-line type motors, illustrating manual start and stop push button. Automatic start and stop switch from float control, pressure control, etc., may be used in lieu of push button

*Consulting Engineer, Chicago, Ill.

with both of these controllers and either one of them will cause the pump to operate. Both of them must be satisfied in order for the pump to stop.

Protecting and Controlling the Boiler-Feed Pump

There is a 10-hp. constant-speed squirrel-cage induction motor of the across-the-line type which drives a centrifugal boiler-feed pump. A float-controlled switch on the receiving tank, similar to those on the vacuum pumps, actuates the motor starter, which is of the same type as those for the vacuum-pump motors.

The flow of water to each of the three low-pressure heating boilers is regulated by a hand valve at each boiler. Should all the valves at the boilers be closed and should the boiler-feed pump still be running, no particular phenomenon will manifest itself, but the pump will continue to run with very little consumption of energy, since its characteristics are that the power demand is proportional to the amount of water which it delivers. If for any reason, however, the motor should be overloaded, the thermal overload relays which are provided in the starter will function to disconnect the motor.

Float Control and Starters for Sump Pumps

The lowest level of the basement is well below the sewer. Seepage and drainage from this zone only flows by gravity into a sump tank prepared for it. No sanitary sewage is handled by these pumps.

Two pumps are immersed in the water near the bottom of the sump tank; their shafts extend to two 7½-hp. motors above the floor line. This installation in duplicate is provided as insurance against flooding the sub-basement in case of failure of one motor, or in case of a load exceeding the capacity of one pump.

The motors are of the vertical type. They are especially designed across-the-line squirrel-cage induction motors, the starting current of which is within acceptable limits. Motors of this design up to 30-hp. in size may be thrown across the line without disturbing unduly the voltage of the line or interfering with other connected loads.

The starters for these motors are actuated by guided copper floats in the water of the sump. These starters are provided also with a selective switch to permit alternative manual push-button operation of the motors.

The float for one pump is set so that the pump will be operated when the water reaches a certain level. The float for the second pump is set to operate the pump if the water approaches a somewhat higher level, since such a higher water level will indicate that the first pump has failed to function or that its capacity has been exceeded. A separate alarm gong is set off by a third still higher float in the tank if both pumps fail to control the situation.

The starters for such a duplex installation of motors generally are contained in a single cabinet where there is provided:

1. Two fused line switches.
2. Two magnetic contactors for closing the line to the motor and for providing under-voltage protection.
3. A double-throw switch for interchanging either motor with either control unit.
4. A double-throw test switch in one position of which the starter is controlled by the float, in the other position of which the motors may be started by a push-button independently of the position of the float switch.
5. Temperature overload relays which disconnect the motor from the line in case of an excessive current demand caused by an overload of the motors.

40-Hp. Motors Have Starters of Compensator Type

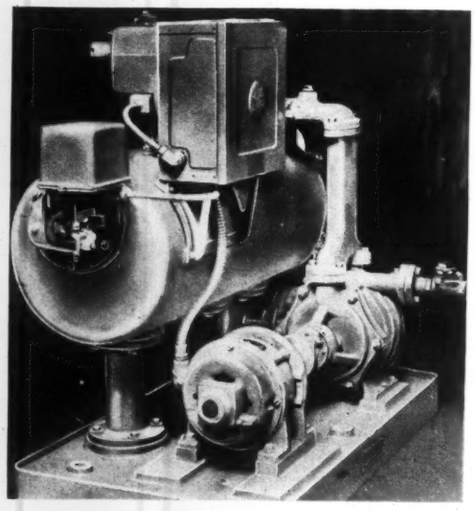
There are two multi-stage centrifugal house pumps in the sub-basement drawing water from the city mains to supply house tanks in the pent house above the 26th story. They are equipped with 40-hp. squirrel-cage induction motors; these are too large to be thrown directly across the line for starting.

These motors, therefore, have starters of compensator type, which cause reduced voltage to be supplied for a definite time during acceleration; then, after the motor has been accelerated, they cause it to be thrown across the line. There is a thermal overload protective device to disconnect the motor from the line in case of excessive current demand.

The starters are actuated by a copper float connected to a chain and counter-weight, and a switch which is mounted on the house tank. The copper float is confined within an 8 in. pipe extending through the top of the tank from a point a short distance above the bottom of the tank. This allows the float to ride at the water level of the tank and eliminates the possibility of its moving violently in case of turbulence or agitation of the water in the tank proper. This pipe guide also prevents the float from drifting within the tank, and sending an erroneous message to the switch.

At the low-water level of the float, the switch is thrown by a trigger on the float chain so as to close, sending energy to the starter for the pump motor. At the high-water level of the float, the switch is opened by another trigger on the float chain; this disconnects the pump motor from the line.

As a further precaution



An electric-motor-driven vacuum pump. The motor starter box is mounted on brackets on the condensate receiving tank. The vacuum switch is immediately behind the starter box and the float switch is at the end of the condensate receiving tank

against overflowing of the tanks (an accident which might occur in case of failure of the pump to stop when the high water level is attained) and as a precaution against a deficiency of water in the house tanks (which might occur in case the water demand from the house tanks exceeds the capacity of the pumps) a separate high-low water alarm is provided in the pump room. This consists of two float-operated switches mounted on the house tank, which, when either float reaches its operating level, will cause an alarm gong to sound in the pump room and a light to glow on a suitable panel—white for low water and red for high water. The alarm gong may be stopped manually by the engineer but the light will glow until the water again returns to the predetermined limits.

Controls for Synchronous Motors

Synchronous motors may be used for driving pumps where power factor improvement is desired. Had this type of motor been selected for the house pumps, the starters would have consisted of devices which include:

1. A fused line switch.
2. Accelerating contactors operating from definite time relays which apply reduced voltage from a transformer to the motor during the accelerating period, and which provide under-voltage protection.
3. Line contactor which closes the motor to the line when synchronous speed has been reached.
4. A field relay-contactor which allows field excitation to be applied only when the motor has reached a speed near synchronism.
5. Temperature overload relays which disconnect the motor from the line in case of an excessive current demand.
6. Line and field ammeters and pilot light are provided for indicating values of current and when the line voltage is on.
7. Rheostat for regulation of field excitation.

Manual Push-Button Switches for Circulating Pumps

There are small centrifugal pumps used to circulate hot water through the piping in the several zones of the building. Each pump is equipped with a 1½-hp. constant speed across-the-line type induction motor.

The starters for these motors are manually push-button operated, magnetic switches with overload protection and under-voltage release. There is a fused line switch built into the same cabinet with the above devices.

Separate Circuit Breaker for Fire-Pump Motor

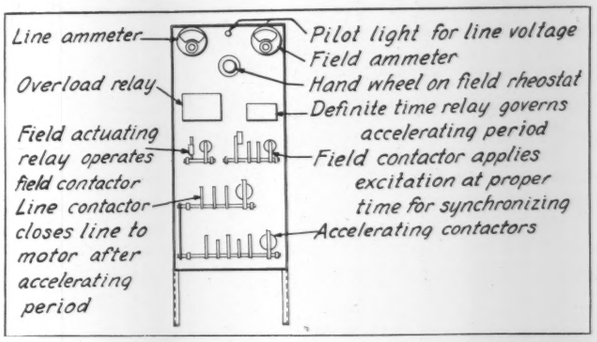
There is a 75-hp. slip-ring induction motor for driving the fire pump.

Power leads for a fire-pump motor come directly from the utility company's service and not through the main switchboard and circuit breaker whence the power service for the remainder of the building is taken. This insures a constant source of energy independent of the motor loads in the balance of the building. However, a separate circuit breaker of the instantaneous type, set for 400 per cent of the full load current, is provided on the incoming power lines to prevent damage to the power company's equipment in case of a short circuit or in case of an excessive overload on the fire pump circuit.

The equipment required for the operation of the fire pump consists of:

1. An air compressor, the motor of which may be operated manually to maintain a predetermined air pressure. The air compressor discharges into a pneumatic tank which serves as a reservoir.
2. A combination automatic and manual starter for the fire pump motor.

The automatic starter is actuated by a pressure-control switch on the dry (air pressure) side of the piping system. This switch is mounted on the starter control panel, with an air pressure pipe connection taken from the air-cushion side of the pneumatic tank. It is set to close the line to the motor starter whenever the air pressure drops to a predetermined value, such as would


Line ammeter
Overload relay
Field actuating relay operates field contactor
Line contactor closes line to motor after accelerating period
Pilot light for line voltage
Field ammeter
Hand wheel on field rheostat
Definite time relay governs accelerating period
Field contactor applies excitation at proper time for synchronizing
Accelerating contactors

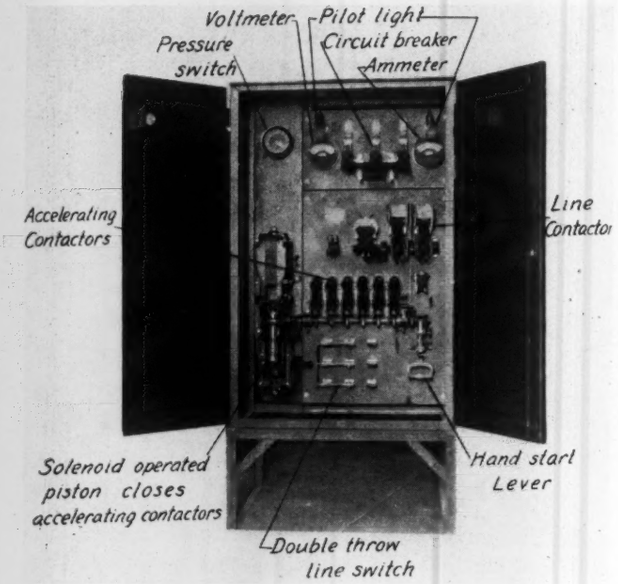
Arrangement of equipment on a synchronous motor starter panel

occur if a cap of the Siamese hose connector at the street level were removed, or if a sprinkler head should open, or if a fire-hose connector inside the building should be turned on, releasing the pressure confined in the system.

The equipment comprising the automatic starter, and its function, is as follows:

1. An unfused main line knife switch serves as the connector to the incoming power leads. This may be a double-throw switch for connecting the motor to either of two power sources.
2. A circuit breaker on each incoming lead is set to trip at a current equal to four times that of full load of the motor.
3. A motor-operated thrustor or solenoid-operated piston and dashpot energized when the pressure switch functions, controls the reduced voltage starting of the motor. This consists of a cylinder containing a piston, to which is attached a shaft on which the contactors for the fire-pump motor starting resistances are attached.

   The fire-pump motor is started under reduced voltage, since placing a motor of large size directly across the line might bring about such a great reduction in voltage to the remaining service leads as to cause all connected motors on the leads to be stopped by their under-voltage release mechanisms. When pressure in the fire piping system is released, the pressure switch energizes the solenoid or closes the thrustor motor circuit, moving the piston and thereby releasing the starting resistance contactors, at predetermined time intervals as the motor is accelerated. After the last of the starting resistances has been cut out, a contactor is engaged to energize a relay which closes the line to the motor.
4. A fuse-protected voltmeter and an ammeter are mounted on the starter panel to indicate line voltage and line current.
5. Two pilot lights are provided, one for indicating the presence of voltage on the line, and the other for indicating



Combined automatic and manual fire-pump controller for slip-ring induction motor

when the full starting operation of the equipment has been completed.

A manually-operated lever is provided in the fire pump control cabinet for starting the pump, as an alternative to the automatic pressure-switch control.

No overload protective device is installed on the fire pump motor. Should the pump motor receive an overload when its service is required, the motor might as well burn out in service in an attempt to supply fire protection as to be thrown off the line by an overload relay and possibly be burned up in the fire.

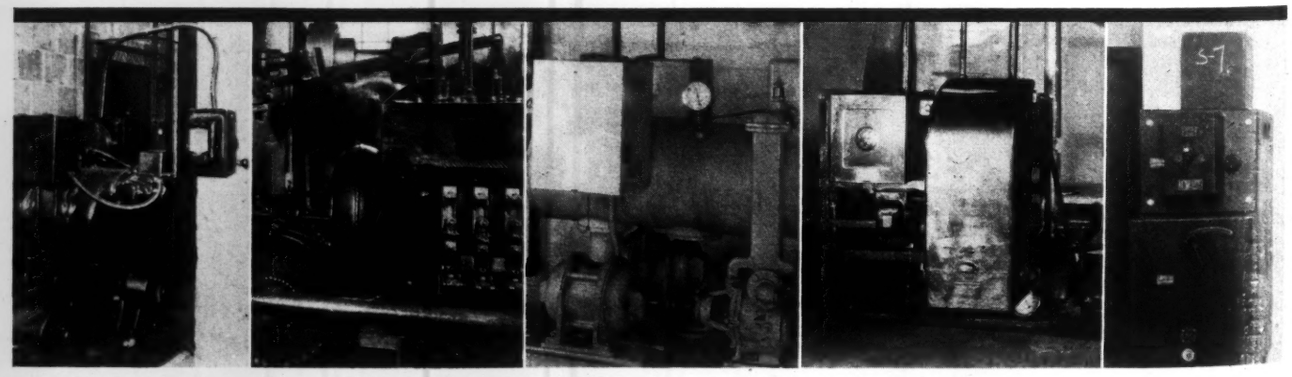
Observe These Points in Connection with Motor Protective Devices

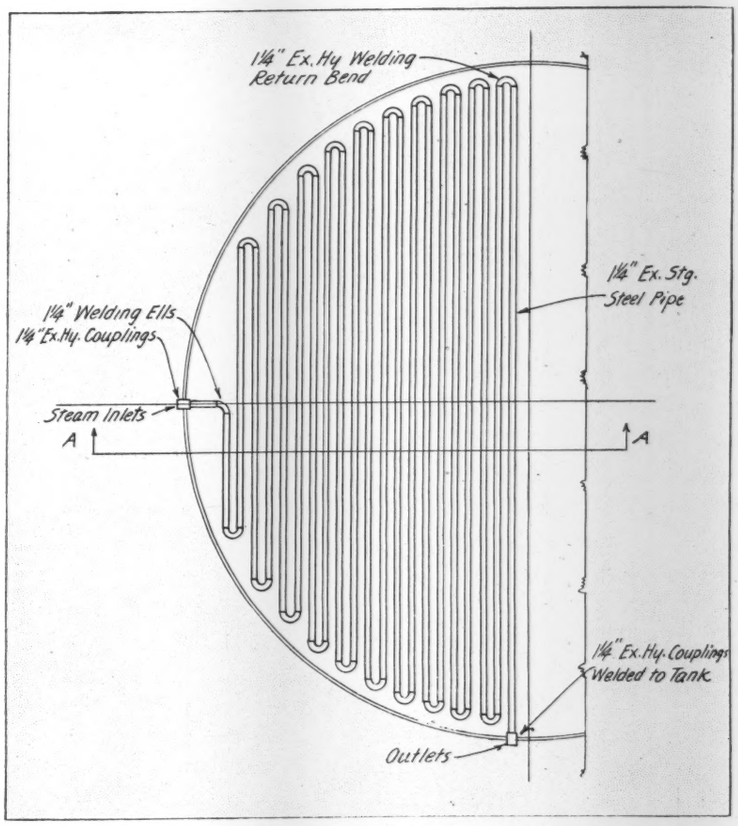
In general, the following facts may be observed in connection with motor protective devices and their use on motor starters:

1. Under-voltage protective devices should be installed on motors generally. These may consist of magnetic holding coils or relays which may:
   (a) Stop the motor under reduced voltage. When the voltage returns to its proper value, the motor again is caused to operate. These protectors are used in connection with starters which are actuated automatically, as by float and pressure switches.

   or which may:
   (b) Stop and release the motor under reduced voltage so that when the normal voltage returns, the motor starter must be energized before the motor can be put back into service. This is a protection against injury to mechanics who might work on the motor without first opening and locking the line switch.
2. Overload protection should be installed on all motors except those for fire-pump service. This may consist of thermal relays for disconnecting the motor when it is overloaded; magnetic relays also may be used for this service.
3. Phase-failure protective devices are installed only for motors which use alternating current and the source of power to which is a poorly-operated local generating station. Usually there are enough protective devices in the generating plant of a modern electric utility company to prevent damage to motors on the outside power lines should one phase of the generator fail. Phase-failure protection is costly and usually is an unnecessary expedient for most installations using public service power.

   If for any reason one phase of the supply to a three-phase motor should fail, the motor would continue to operate on single phase only, as the two remaining line leads would function as conductors for flow and return of one phase only. The current, however, would increase to 1.73 times its three-phase value on a single conductor. This amount of current would cause the thermal overload relay to function and would disconnect the motor. Thus a thermal overload relay also serves as a phase-failure protector.





LIMITED SPACE WATER-LOGGING

Complicate Steam-Coil Installation

Instead of one continuous coil, three separate coils, with individual inlets and trapped outlets, welded, meet the problem in a chemical process plant

. . . . . *By* G. W. Hauck*

A CHEMICAL process plant required some steam-heating coils; because of limited space and other requirements, considerable thought was given to the design. Believing that the experience gained will prove of value to many such plants, this article describes the solution of the problem.

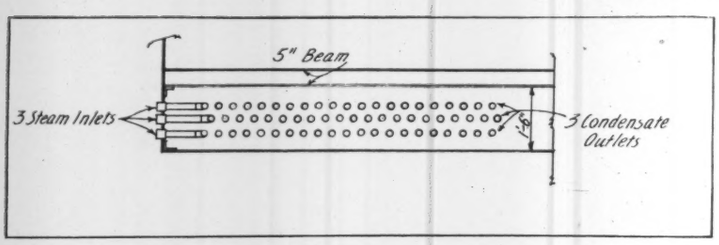
Steam-heating coils were required in a tank 16 ft. in diameter, with a small access door located at the right on the center line opposite the steam inlets shown in Fig. 1. This access door is not large enough to permit complete assembly of the coils outside and so all piping had to be partially assembled outside and final assembly completed inside the tank. The nature of the process required coils only in approximately one-half of the tank, located in a space between the bottom of the tank and the bottom of 5-in. beams. This space is 1′ 6″ in height as indicated in section *A-A*, Fig. 2. In this space of approximately 1′ 6″ x 8′ 0″ it was desired to install the maximum practical amount of coil heating surface.

In previous installations of a similar character in this plant, considerable difficulty had been experienced through action of the chemical at the joints, which, with the movement of the piping, in time developed leaks, not only causing loss of valuable steam but having a deleterious effect on the product in the tank. In this case it was decided to use welded joints throughout. The coils are of 1¼ in. extra strong pipe with 1¼ in. extra heavy iron pipe size welding return or 180-deg. bends. Extra heavy iron pipe size 90-deg. welding ells were used at the first turn beyond the steam inlets. At the three inlets and outlets, 1¼-in. extra heavy hydraulic couplings were welded into the tank shell. The terminal ends of the pipes were screwed into these couplings, and, after being screwed up, were welded to seal the joints.

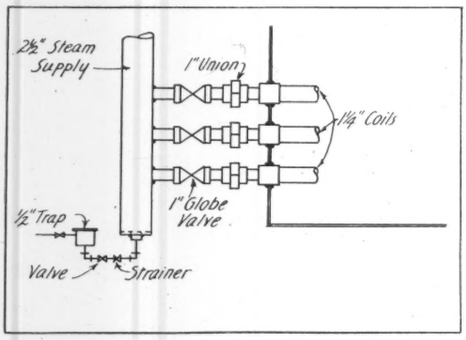
Because of the small scale of Fig. 1, there is shown only the plan view of a single

Fig. 1—Because it was necessary to get the maximum amount of steam coil heating surface in this process tank, and prevent possible leaks due to movement of the piping, considerable thought was given the layout. Above is a plan view of one of the three coils in the tank.

*Cr[illegible]e Co., Cincinnati, Ohio.



coil. Three of these coils were installed and the pipes staggered in the alternate tiers as indicated in section *A-A*, Fig. 2. It is believed that the method of staggering the pipes produces better results in heating than the method of placing the pipes of each coil directly above the other, in line vertically. Steam at approximately 80-lb. pressure is used for heating, and this, as well as the nature of the process, necessitated the use of pipe coils; the arrangement used provided the maximum of piping possible in the available space. It will be noted that the angles used for supports in the tank construction reduce the piping space of 1′ 6″, so that the three coils had to be installed on approximately 3½-in. centers vertically.



In previous installations, the piping had been assembled in one continuous coil with a single inlet and outlet. Operating experience had proved this inefficient, because rapid and uniform circulation of the steam was not obtained at all times. In starting up cold, considerable condensation of the steam occurred and the lower part of the coils would be more or less water-logged. This difficulty was eliminated by using the three separate coils, each with individual steam inlet and individual trapped outlet. As the process continues, less heating surface is required as the material is heated; with the arrangement shown flexible control is possible, as single coils can be cut out of service by the operator from time to time. Installation of the separate coils, rather than one continuous coil, also simplified the installation and welding of the piping.

A detail of the steam inlet connections is shown in Fig. 3. A 2½-in. steam supply line is formed into a header by welding the three 1-in. branches into the 2½-in. piping. As this 2½-in. steam supply line is about 60 ft. in length, it is drained at the low point through a ½-in. steam trap. A piece of plate is welded in the bottom of the 2½-in. pipe and in this plate an extra heavy coupling is welded for the drain connection. The 2½-in. supply header was carried about 12 in. below the lowest 1-in. branch to the coils, providing a drip pocket to insure proper drainage of the supply header. The three supply branches were made 1 in. in size to permit close coupling of the valves and unions on 3½-in. centers and to permit turning of the union rings. 1¼″ x 1″ face bushings were installed in the 1¼-in. couplings welded in the tank shell. The 1-in. steam supply branches are amply large so it was not deemed necessary to have these branches 1¼ in., the full pipe size of the coils. A globe valve in each of the three branches permits individual control of the coils.

Fig. 4 shows a detail of the condensate outlets. Each coil is provided with an individual trap of small size, but sufficient capacity for ordinary service. A gate valve before each trap inlet and a similar valve on each outlet permit the ready removal or repair of any trap. These valves also permit shutting off the condensate line when any coil is cut out of service. The strainers shown in each trap line serve a two-fold purpose. They not only keep sediment and scale out of the trap valves, but the drain valves on each separator can also be used to assist in relieving the condensate when first starting up cold under heavy steam load, when there is considerable condensate and its quick removal is desirable. This process plant is such that there is a surplus of water for boiler feeding and so some wastage at periods is not objectionable. After the material begins to absorb heat, these valves on the strainers are closed and the traps can handle all condensate. It is only for occasional short periods that these strainer drain valves are used to relieve the coils of excess condensate.

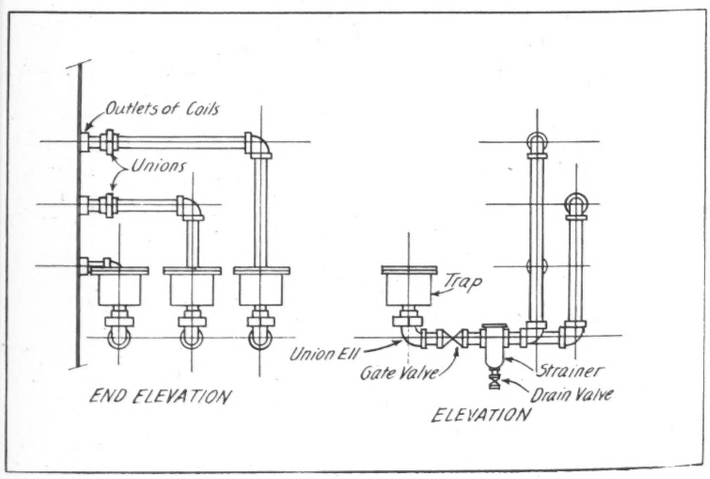


Fig. 2 (Top)—Section A-A of Fig. 1, showing how the three tiers of coils were staggered in the 1′-6″ space available, in order to increase the heating effectiveness. Fig. 3 (Center)—Detail of steam inlets. As the steam supply line is about 60 ft. in length, it is drained at the low point through a ½-in. trap. Fig. 4 (Bottom)—Detail of condensate outlets; each coil is provided with a steam trap. Each trap discharge has a gate valve, union and check valve connecting to a short header

Drying—an Essential Process in Industry

The proper control and application of temperature and air circulation required in many processes are made clear in this article on leather drying . . . *By* Walter L. Fleisher*

The whole process of leather making is a colloidal chemical problem, and the drying of leather for its final handling is only one of the steps in the coagulation of a colloidal material. In order to treat leather it is essential to have an evenly-hydrated material so that the action of the solutions essential to its curing and waterproofing may be evenly accomplished. When this curing and waterproofing of the minute cells has been accomplished, it is necessary to carry away the excess water of hydration without disturbing the chemical nature of the collagen; for this reason careful drying has always been an essential part of the preparation of all leathers.

To the air-conditioning engineer the hydroscopicity of finished leather is extremely interesting because of the use of leather in manufacturing processes. After it is tanned and finished and dried, leather is greatly affected by the moisture content of the surrounding atmosphere. In the manufacture of shoes, for instance, care has to be taken in the selection of the kind of leather and the method used in its tanning, in order to eliminate stretching as much as possible from the sole, and to give the proper stretch to the uppers.

THE thermodynamics of leather drying are practically the same as the thermodynamics of any other drying. Air, at a definite dry-bulb temperature and dewpoint, has a capacity to absorb a certain additional amount of moisture if not saturated at the beginning, and its absorption power is, to a large extent, dependent on the dry-bulb temperature. As this air comes in contact with the damp leather, two things happen. In the first place, the air becomes more nearly saturated and, secondly, the dry-bulb temperature drops, due to the loss of heat required to evaporate the water which is absorbed as a gas by the air.

Absorption Capacity of Air Is Reduced

In practically all cases, there is a wide spread between the moisture content of the air and saturation even after the air has passed over the leather, because the more nearly air approaches saturation the less avidity it has for extracting moisture from the leather with which it is brought in contact. There is also always a tendency for the dry-bulb temperature of the air to drop due to its moisture pick-up and thus the air becomes less capable of absorbing moisture. The transmission losses through the sides of the dryer and through the out-leakage of air, both in the entering and delivery end of the dryer, also tend to lower the dry-bulb temperature and thus lower its capacity.

20 Per Cent of Air Lost by Leakage

The air which has leaked out has to be replaced by fresh air, either from the room itself or from outdoors, whichever is more suitable and economical for the purpose. This fresh air, as well as the air which has been used and which is to be re-used, must be heated or reheated in order to have absorbing power. About 20 per cent of the air circulated, no matter how efficient the dryer, leaks out and must be replaced with other air.

Methods of Creating Conditions Conducive to Drying

In almost every drying proposition, particularly in leather, the methods of creating conditions conducive to successful drying are practically as follows:

Air is heated to the maximum temperature that the leaving material will stand and this air at that temperature, usually around 150 F in leather, sometimes higher, has a great and rapid power of absorption. This air, which is to be heated to 150 F, is only partially saturated at its entering temperature of 80 F dry bulb, being, let us say, at 50 per cent relative humidity and containing only six grains of moisture per cubic foot. When this same air is heated to, say, 150 F, it has a theoretical absorption capacity of about 70 grains per cu. ft. and therefore takes up rapidly whatever moisture is attracted to the surface of the leather.

*Consulting engineer, New York City.

It is customary to divide a dryer for leather into from four to six drying stages. Therefore, if 150-F air is delivered over the surface of the leather at the leaving or finishing stage and absorbs two grains of moisture per cu. ft., the temperature of the air would drop due to evaporation approximately 16 F and due to radiation probably another 16 or 20 F. The air could then be re-used in a state further along in the dryer, possibly picking up two grains more and dropping to possibly 90 F.

Here, with ten grains of moisture per cu. ft. and only 90-F dry bulb, the air would have a relative humidity of about 70 per cent and would do very little work; the leather would hold its moisture due to its vapor pressure with a fair amount of avidity unless the air was reheated. This reheating is usually done, and the air brought back to, say, 120 F at this point, but without removing any of the moisture in it. This air would then, when passed over another stage of the dryer, be able to absorb additional moisture. It could then be reheated by steam coils or other means, or could be allowed to heat up the entering leather so that the moisture throughout the pores of the leather would be brought to a position of equilibrium or vapor pressure where it could be readily absorbed in the next stage.

By this method the leather would be kept in good condition until the vapor pressure throughout the entire mass was in partial equilibrium and without any tendency of the outside pores of the leather to case-harden.

With six grains per cu. ft. original moisture and two grains pick-up through each of two additional stages we have only ten grains per cu. ft., which would seem to be a very small number of grains when the air at 150 F has a capacity to absorb 70 grains per cu. ft. However, it must be remembered that at the point where the 150-F air strikes the leather, the leather is practically dry and it would take almost a baking process to get more than the two grains before mentioned out of the pores of the leather at this stage.

The probabilities are that in the last stage, just before the leather leaves the dryer, not over one grain or half a grain per cu. ft. is extracted, whereas in some central stage, with a lower temperature and more moisture per cu. ft. in the entering air, the leather is in such a condition of vapor equilibrium that it readily gives up moisture and two or three grains of moisture can be extracted at that particular point.

Time Element Must Be Considered

There is something else to be considered also in the extraction of moisture, and that is the time element. All drying processes are a function of this time element and in a mechanical dryer of the nature of a leather dryer, where the air usually passes from the top of the dryer down and out at the bottom, the length of time that the leather is in contact with moisture-absorbing air is very short. Of course, each propeller fan located over the top of a dryer delivers practically a continuous stream of air in approximately vertical lines from the top to the bottom of the dryer and the leather, in progressing through the dryer, passes through the zone of each particular fan for the length of time that it takes one piece of leather to go through the distance occupied by the diameter of the propeller fan. Of course the velocity with which the air is passed over the surface of the leather would also affect the drying time and the case hardening as well.

By long practice and experiment and experience the engineer would have to decide what temperatures were best fitted for the leather at various stages of the drying process and in what period of time he wanted his leather dried. Then, knowing the initial weight of the wet leather as it was brought to the dryer and the percentage of moisture which it was desirable to retain at the finish of the drying, it would be easy to calculate the amount of heating surface required to bring the air up to the temperature required for the work. The calculation of the amount of air and the time required to remove the moisture is also a simple problem in primary thermodynamics. He could also, from the above data, figure the number of stages desirable for the particular process and from the results obtained could calculate the number of pounds of steam and the amount of electricity that would be required to do the drying.

A Typical Example Worked Out

As a particular example, assume that a man wanted to dry 2,000 skins an hour from which 2,000 lb. of water were to be removed. The calculation would resolve itself into how much air and at what temperature it would have to be delivered to absorb 2,000 lb. in that period. Assume that over a four-stage dryer eight grains of moisture per cu. ft. could be picked up and that fourteen million grains had to be absorbed in an hour. This divided by eight, the number of grains which were to be absorbed per cu. ft., would give approximately 29,000 c.f.m. of air to be circulated in order to do the work. If this were done in three stages, it would mean that three 10,000-c.f.m. fans would be used, or probably slightly larger fans in order to provide a factor of safety. In a case of this kind 15,000-c.f.m. fans would probably be installed. Steam coils would have to be provided to heat 15,000 c.f.m. of air from the room temperature, say 70 F, to 150 F, and then other coils to heat air partially saturated at, say, 120 F back to 130 F and then from 90 F back to 110 F, at each stage the air holding slightly more moisture.

From ordinary blast heating practice the engineer could figure the sq. ft. of coil necessary and the amount of steam condensed in accomplishing these results.

If the amount of moisture to be removed in an hour is too great for the leather itself to stand, it is customary to build a dryer of twice the capacity and allow the leather to remain in the drying chamber over a period of two or three hours, or longer. In

such a case, the calculations for absorption are simply proportioned to the number of hours in which the drying is to be done, additional consideration being given to the transmission losses over the additional period.

In leather drying, where the leaving end is brought to a high temperature or a temperature higher than the ordinary exhaust steam that a manufacturing plant would produce, it is often customary to have some of the coils operate with high-pressure steam and the others under the exhaust-steam-pressure of the plant.

The fans which create the air movement so essential to proper and scientific drying are usually of the propeller type, as the resistance in dryers of this type is not great, and a type which will handle large volumes of air over rather large areas without the necessity of duct work is usually preferable. Where a series of propeller fans are used, they may either be driven by direct-connected motors, or by belt drives allowing for simple variations in the speed of the fans. The series of fans can be driven from one shaft.

Drying Conditions Should Be Automatically Controlled

The methods of controlling drying temperatures and relative humidities for proper drying employed in other drying or in air conditioning apply equally to leather drying. Dry- and wet-bulb thermostats installed at various portions of the dryer control diaphragm valves on the steam coils to provide the heat required, or introduce additional fresh air to lower the absolute humidity and thus slow the case-hardening which may occur with too rapid drying.

As has been stated, when air absorbs moisture a drop in temperature takes place and heat must be applied to replace the sensible heat lost. If the drying is sluggish and if the dryer is not filled to capacity, the amount of moisture absorbed by the air is very often less than that calculated and consequently too high a temperature may result; thus conditions inside the dryer should be automatically controlled.

Quite often leather drying is done in cycles of high and low temperature; this can be controlled by means of automatic time mechanism applied to the heating coils.

The conveyors in a dryer are often controlled by variable speed motors to give different drying times; this can be done either by automatic means or by hand control.

The amount of power required for driving a fan is a simple matter of calculation and the amount of power required for driving the conveyors is a matter of experience and practice.

If part of the partially saturated air at the entering end were to be re-used and mixed with a certain amount of air from the room, a slight complication in the problem would arise, but one that would be perfectly simple for the average engineer to work out. As one engineer has remarked, "Each leather problem is practically a new problem." The method of handling the goods, the intermediate dampening, the amount of moisture left in the leather due to a slightly different chemical treatment, the different colloidal content of the leather at different plants due to the different methods of curing and operating, bring up problems that no general description of leather would enable one to cope with.

Saving Money in Heating Water for Use in Buildings

. . . . . *By* L. K. Ferris

IN HEATING water for domestic use in office buildings, hotels, hospitals and institutions, material economies may be secured by careful control.

In large office buildings the use of hot water may be 4 to 6 gal. per occupant per working day, including the use for cleaning and janitor service. Heating this water requires an amount of heat depending upon (1) the temperature of the cold water supply, (2) the temperature which is required at the point of use, and (3) the loss of heat from the water occurring during its passage through the piping system.

Several sources of heat are available to raise the temperature from that of the cold-water supply to that required for use. In the northern states the annual average temperature of cold water supply is about 50 F. The water should be taken from the cold-water system at some point where the cold water is usually several degrees warmer than the temperature of the actual supply from the city or other source. In many cases the few degrees gained this way represent 6 or 7 per cent of the total temperature increase desired.

Keep Maximum Temperature Low as Possible

The final temperature to which the water is heated is often greater than necessary. In office buildings and hotels a maximum temperature of 140 F is usually satisfactory. In hospitals it may be necessary to carry a higher temperature. In any case, a check should be made to determine the minimum temperature that will satisfy the requirements of all users. A schedule of reduced temperatures may be established for operation during hours when lower temperatures are satisfactory. It is often only for the last increases of temperature

that any great quantity of steam or fuel is required to obtain the desired final temperature. In many cases it will be found possible to recover otherwise wasted heat to maintain the hot-water supply at 125 F at all times and at certain times considerably higher. By establishing maximum water temperatures as low as possible heat is thus conserved.

Sources of Waste Heat

Most modern buildings have some refrigerating equipment operating practically the year 'round and requiring condenser cooling water. In many cases the volume of cooling water for the refrigerating plant is far greater than that required for the hot water supply system. The refrigeration plant probably will operate with an outlet water temperature from 10 to 20 F above the city supply temperature; the piping may be arranged so that the hot water system is supplied by water from the condensors of the refrigerating plant.

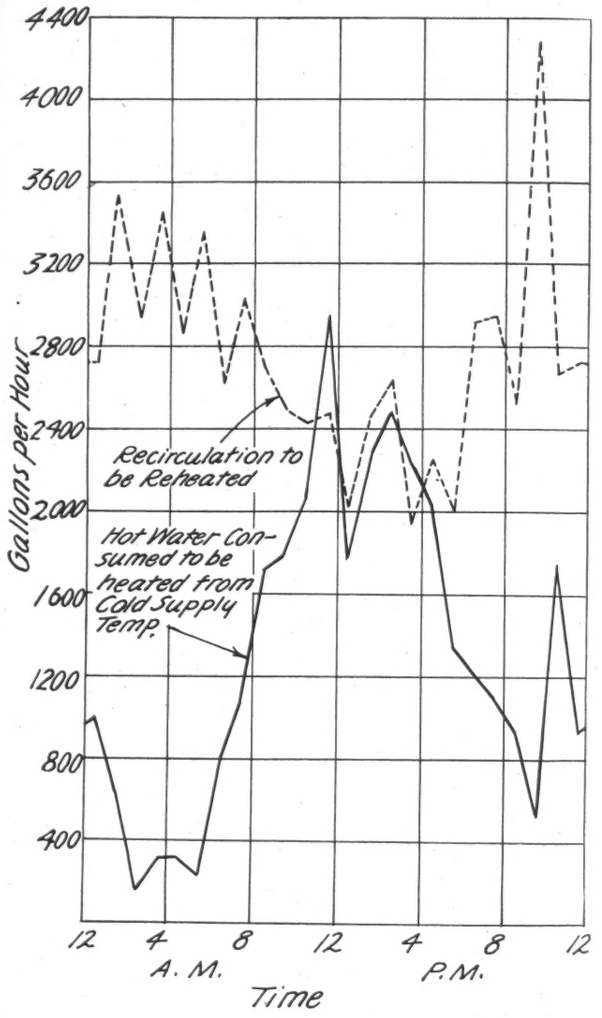
It is common to provide means for recovering heat from the steam-heating system condensate returns. The temperature to which the hot-water supply may be heated by this means depends on the efficiency of operation of the heating system. In a well-operated heating system the returns may range between 100 and 130 F except on extremely cold days when the heating system is operating at full capacity. During the heating season—with properly selected heat-exchange equipment—it should be possible to raise the temperature of the hot-water supply to within a few degrees of the temperature of the heating returns. This, of course, will depend on the size of the heating load in proportion to the hot-water consumption.

Still additional heat is usually available above that which can be obtained from heating-system returns. Drips from high- and intermediate-pressure steam lines, condensate returns from kitchen equipment, laundry equipment, etc., may be piped to heat exchangers. In many plants exhaust steam which would be wasted in non-heating seasons is available for the water-heating requirements. In some plants exhaust gases from diesel engines provide heat available for water heating.

Heat Required for Heating Up Return Water Is Important

In installations of any size a pump is used in order that circulation will be rapid enough to prevent excessive drops in temperature between the heaters and the point of use. In addition to the heat required for heating the water actually used, heat must be supplied to make up the losses in temperature of all the water which is returned to the heaters from the circulating system. These losses depend on the length and size of the piping system and the efficiency of the pipe covering. The heat necessary to provide for these losses depends on the ratio of the quantity of water used to the quantity returned and the temperature range of heating compared with the loss in temperature of the return water. In an ordinary installation the heat for reheating this return water is an important part of the total heat required.

The importance of the reheating of the return water is greater than at first seems apparent since the heat required for this purpose must all be obtained from some source at practically the same temperature as the desired final water temperature. Heat at this temperature in many cases is not available from heating-system returns and other waste heat sources; as a result reheating the return water may require more live steam than that required for heating make-up water.



Reducing Circulating Losses Opportunity for Saving Heat

Considering this it is seen that any reductions which can be made in the circulating return losses are very important. These losses may be reduced by (1) use of as small-sized pipe as possible consistent with reasonable losses of pressure in order that the radiating pipe surface will be a minimum; (2) use of efficient pipe covering; (3) use of as short a circulating system as possible; (4) maintenance of as low maximum water temperature as possible.

Data on One Building Show Need for a Study

It will be seen from Fig. 2 that the temperature of water supplied to heaters varies considerably above the temperature of the cold water supply, due to the distance

Fig. 1 (Left)—Hourly consumption of hot water in office building having occupancy of about 6,000 and hourly return water which must be reheated to replace line losses. Data represents average week day in April

Fig. 2 (Below)—The temperature schedule calls for 140 F during the day and 130 F at night; the latter might be further reduced to effect economy

Fig. 3 (Right)—Heat required hourly for water heating

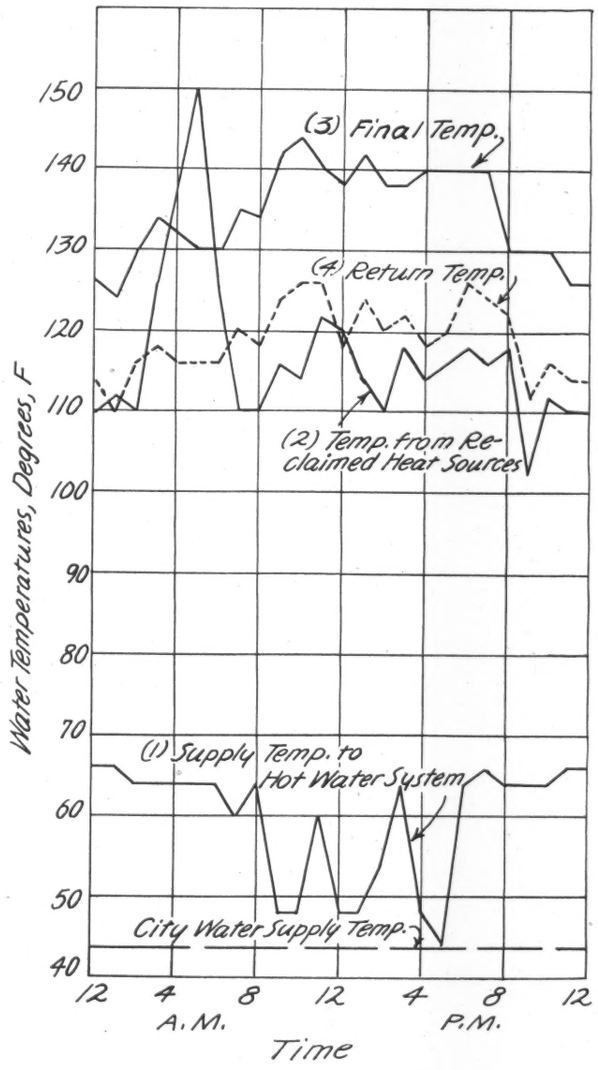


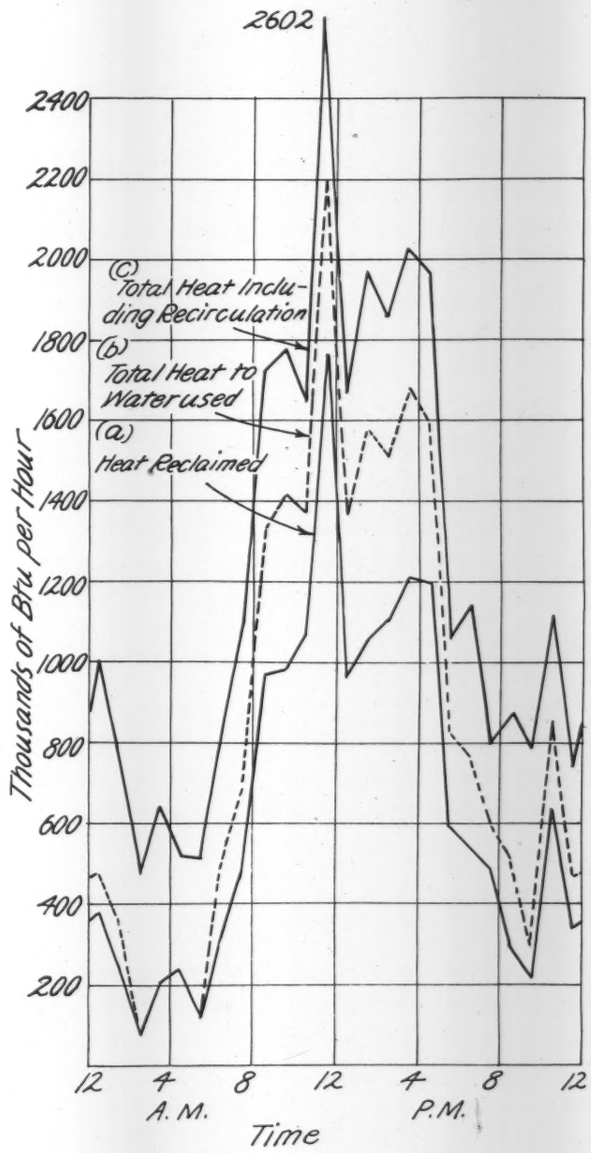
Table 1—Heat Supplied to Water from Various Sources in an Office Building

	Btu per lb. water used		
Month	Jan.	Apr.	July
Temperature increase required, F	90	80	55
Btu from kitchen condensate	13	12	11
" " high pressure drips	14	15	18
" " heating system returns	43	32	0
" " live steam for water consumed	20	21	0
" " exhaust steam for water consumed	0	0	26
" " live steam for reheating water returned	30	32	35
Total Btu required per lb. used	120	112	90
M gallons used per month	840	780	540

from the city supply to the heaters. In this installation the supply to the hot-water system comes from the tanks of the high-pressure cold-water supply and also from condenser cooling water from a refrigeration plant. Heat is reclaimed (a) from high-pressure steam traps draining approximately 1,000 ft. of 6-in. and 8-in. steam line operating at 125 lb. per sq. in. gage; (b) from returns from steam cooking equipment in a large kitchen; and (c) from condensate returned from the building heating system.

Fig. 3 shows the heat required hourly for the water heating. The table below shows the heat supplied from various sources per lb. of water consumed for various seasons.

The data given herewith are only those obtained from tests made in a particular case. They are not offered as an indication of perfect operation. Each building has some peculiar condition differing from that of other buildings. The different seasons of the year will govern the limitations of control. The figures indicate that a study of conditions in each case will disclose the possibilities for improvement in operation and saving in heat.



Heats School With Two Oil-Fired Furnaces

••• Chicago's Board of Education Engineers design a forced-circulation warm-air job for one of its schools, study its operation, test its performance

*By . . . . . John Howatt**

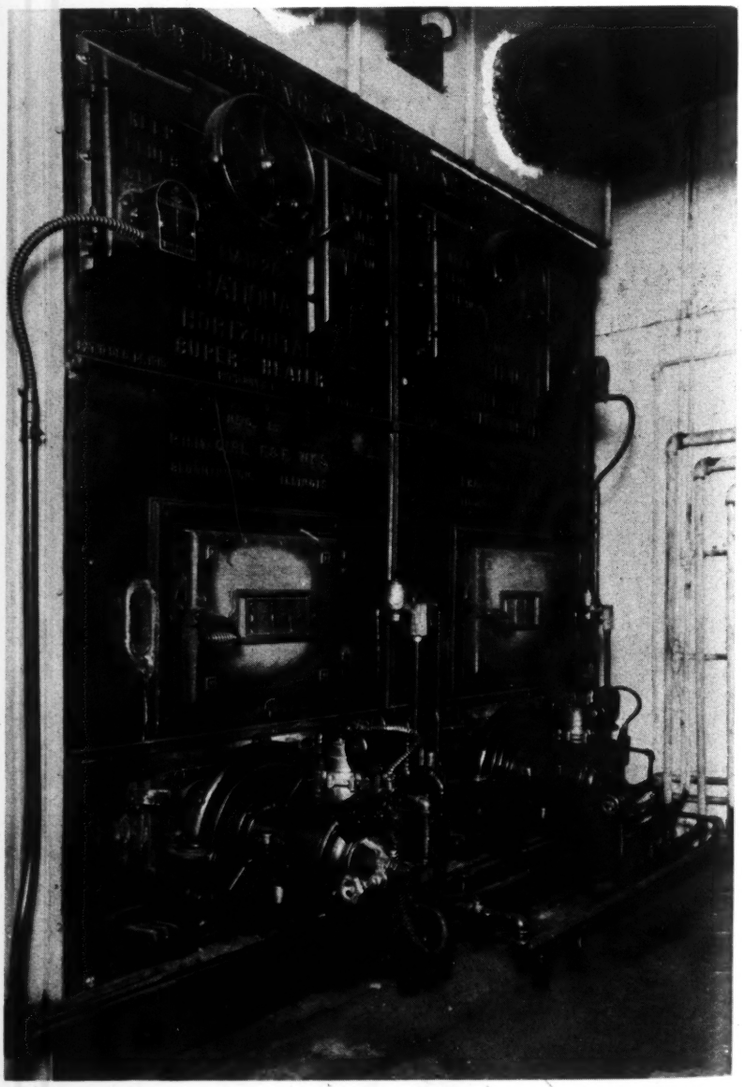
THE latest plan of heating and ventilating Chicago elementary school buildings provides in the main for a fan-blast system with low-pressure fire-box steam boilers with motor-driven fans forcing air for heating and ventilation through tempering and heating stacks and by individual metal ducts from the tempered air sub-corridor chamber vertically to classrooms, with a vent duct from each room up through the attic and out through roof vents, with the usual provisions for either partial or entire recirculation of air when the building is to be heated during non-occupancy.

In 1931, the board of education erected a two-story school of eight rooms, known as the Bret Harte school, to provide accommodations for pupils in a small section isolated by railroad tracks, boulevards and parks.

When the Bret Harte project was considered it was decided that in order to reduce operating costs it would be advisable to install a direct-fired fan-blast furnace system instead of the usual system. Because of limitations in coal and ash storage space and the character of the district in which the school is located the furnaces are oil fired. The system consists of two direct-fired fan-blast furnaces with a motor-driven blower taking air from a fresh air shaft, forcing it around the furnaces and through individual ducts to each room in the building with an intermittent oil burner of the rotary type in each furnace designed to burn 18-20 gravity oil.

The heating load requirement was calculated in accordance with rules given in the *A. S. H. V. E. Guide*, based upon a 15-mile wind and —10-F temperature outdoors with a +70-F temperature throughout the building; sufficient surplus heating capacity is provided in the furnaces to take care of the heating-up load in the morning in addition to the heat required to take care of the heat losses due to the infiltration of air and to the radiation from the building and the heat loss in the air leaving the vent stacks.

*Chief engineer, Board of Education, City of Chicago.



The two direct-fired fan-blast furnaces are equipped with intermittent rotary-type oil burners. An average of 17.0 lb. of oil are burned per day per degree the temperature is below 70 F

The second-floor plan of the building is shown in Fig. 1. The quantity of air required is based upon 25 c.f.m. per pupil in each room as required by the Chicago ordinances. It has been found that under the most severe conditions this quantity of air will take care of the heat losses in the most exposed standard classroom at a —10 F outdoor temperature when the air in the duct leading to the room is heated to 120 F. A selection of the proper furnace for the job was then made, using experience and manufacturers' guarantees as guides.

Clauses in the specifications covering the furnaces were as follows:

"Furnish and install two ——— furnaces with a total of not less than 700 sq. ft. of heating surface. The furnace shall be of cast iron throughout with cast-iron flues extending from front to rear with a horizontal fire travel of not less than 30 ft. before gases enter the smoke pipe. The fire boxes shall have renewable cast-iron liners extending full length along both sides of the grate from front to rear. Furnaces shall be installed by experienced mechanics. The parts must be securely bolted together, with furnace cement in all joints and with bolts located out of contact with fire.

Furnaces shall be encased in steel casings. Casings shall consist of panels constructed of No. 22 gage copper-bearing metal sheets, spaced 2 in. apart and filled with 2 in. of asbestos cell board. Panels to set in "T" iron frames and finished with sheet metal cap to cover "T" bar and 2 in. of panel edge."

Reducing Heat Loss Through Ceiling

Vent ducts leading from the different rooms are terminated through screens in the attic space. From these vents the air discharged from the rooms finds its way through the attic out through a vent tower at one end, the openings into this tower being pneumatically controlled by louver dampers from the janitor's quarters. In this way the air from the room vents is used to heat the attic and the heat loss from the ceiling of the top floor rooms is materially reduced. The building is fireproof throughout, so there is no fire hazard.

Attempt was made to provide a supply of so-called tempered air by taking a large duct out of the bonnet at one side and bringing it back to a chamber from which air can pass to the individual ducts through the mixing dampers. A thermostat is set to maintain this temperature at 65 F by controlling the position of the damper into the chamber into which the blower taking outdoor air discharges. The sketches in Fig. 2 indicate this arrangement.

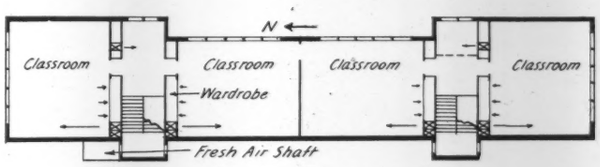


Fig. 1—Plan of the second floor of the Bret Harte school. Air supply is based on 25 c.f.m. per pupil as required by ordinance; this quantity will take care of heat losses from a standard classroom under the most severe conditions when air in the duct to the room is at 120 F

The control of the temperature and supply of tempered air cannot be considered perfect. The adjustment of the relative positions of the two dampers controlling the temperature of the air in the tempered air chamber, the one damper admitting hot air from the bonnet and the other cold outdoor air from the fan blast chamber, must be too carefully made and maintained for a practical satisfactory system. In another of our small school buildings heated by a fan-blast furnace system the tempered air supply is provided by extending a return duct from the hot-air bonnet over the furnaces back to the air intake side of the blower, the amount of such return air being controlled by a damper, the position of which is regulated by a tempered air thermostat set at 65 F. This method avoids, to a large extent, stratification of the air and makes it impossible for drafts of cold outdoor air to be blown up through the ducts and into the rooms. The objection to this method is that a blower approximately 50 per cent larger than is needed to handle the c.f.m. of new outdoor air required must be installed because of the quantity of air that must be handled twice by the blower. This results in a somewhat larger initial installation cost as well as a larger operating cost due to the additional power required by the motor.

Humidification of the Air

Humidification of the air is provided by four cast-iron pans with a total evaporating surface of 1800 sq. in. set in the hot-air bonnet over the top of the furnaces; the water-line in the pans is kept constant by a tank and ball-cock connected to the city water system. A test indicated that with an outdoor temperature of 26 F and a relative humidity of 95 per

SAYS THE AUTHOR

Further study and experimentation should be made on fan-blast furnace type of equipment for school heating to obtain a more satisfactory method of providing a tempered air supply. When direct radiation is not used in conjunction with this type of equipment, room temperature control must depend upon accurate control of duct temperatures.

When two furnaces are set in a battery connected to one common furnace bonnet chamber from which ducts lead to different classrooms, provision should be made so that the air temperature throughout the bonnet space will be fairly uniform whether one furnace or two furnaces are in operation. This is essential to provide flexibility and efficiency in operation. It should not be necessary to operate both furnaces in mild weather to provide a warm air supply for all ducts regardless of point at which they leave the bonnet chamber.

Resistance through gas passages of most furnaces of this type is comparatively low so stack losses will be high unless the stack draft is checked. It is desirable that in furnace installations using oil burners that some form of balanced check damper in the breeching be provided.

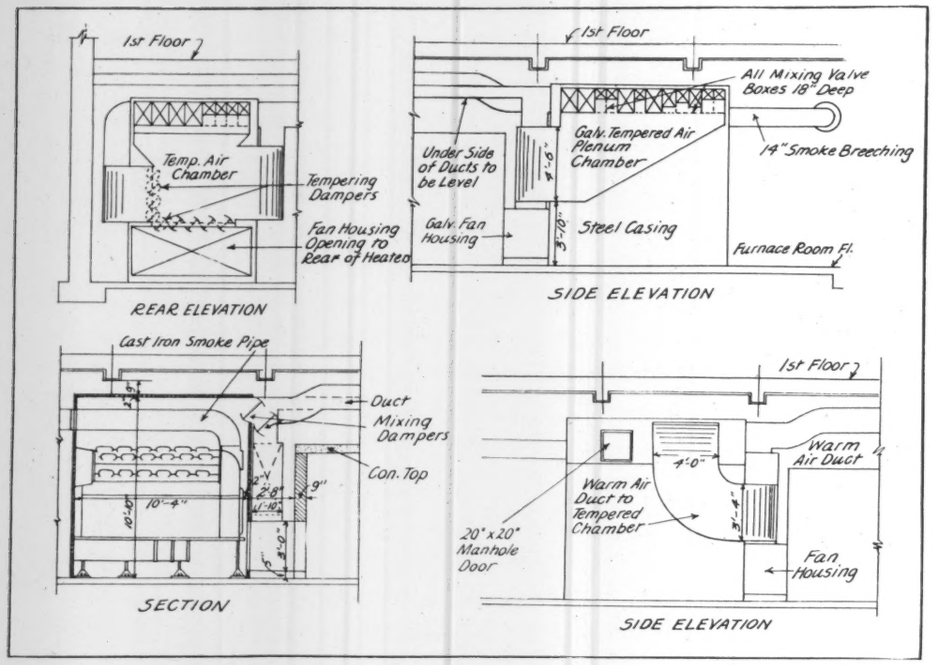


Fig. 2—Attempt was made to provide a supply of "tempered" air by taking a large duct out of the bonnet at one side and bringing it back to a chamber from which air can pass to the individual ducts through the mixing dampers; this arrangement is indicated by these sketches of the furnace

cent, with a room temperature of 68 F, there was maintained an average relative humidity of 28 per cent. Therefore, 0.074 lb. of water was being added to each 1,000 cu. ft. of air. The humidifiers would have been more effective if a pre-heating coil had been provided.

Automatic Regulation of Temperature

Automatic temperature regulation (which has been found to be most important and difficult in this type of installation) is provided by room thermostats and pneumatically-controlled dampers at the furnace chamber. A thermostat set in the tempered air chamber controls the temperature of the air in that chamber by regulating the relative quantities of cold and hot air in the mixture, and a thermostat in the hot-air bonnet cuts off the oil burners when the bonnet temperature reaches 150 F and starts them when the temperature falls to 120 F.

Test Run to Determine Performance

Tests of plant and building performance were made last February and March.

An attempt was made to determine the efficiency of the furnaces with the heating and ventilating system operating under normal conditions in which room dampers are controlled by room thermostats, the volume of air delivered to the rooms being constant but the temperature of the air in the ducts changing to meet the heat demands in the rooms. It was soon found that while considerable interesting information was obtained from the observations it could not be used in determining a furnace heating efficiency because the rapidly changing temperature conditions in the different ducts made any summation of the heat in the air in the different ducts through any definite period of time practically an impossibility. A second test, therefore, was conducted in which the thermostatic control of the dampers was eliminated and all dampers were set so all ducts leading from the furnace chambers received hot air from the top bonnet chamber only. Inasmuch as the oil burners on the job are of the intermittent or on-and-off type, the burner on one furnace was shut down and the air supply around the furnace casing closed off. Under this condition the oil burner on the furnace left in service would be required to operate continuously in order to maintain bonnet chamber temperatures and in this way a comparatively fixed condition could be maintained throughout the period of the test.

Stratification in the Ducts

As was to be expected, a certain amount of stratification of air in the ducts and even in the bonnet chamber of the furnace was present, as indicated by differences noted in temperature in moving thermometers through comparatively small distances. It was found, for example, that there was some difference in temperature observable in different points in the cross-section of room ducts leading from the furnace which would lead to error in the test calculations. The duct thermometers were therefore set with their bulbs in the air stream at the center of the duct; this was considered representative of the temperature of the air across the entire duct area. These temperatures were used in calculating furnace output. Differences of as much as 20 F were observed in temperature of air in different parts of the furnace bonnet and, as was to be expected, difficulty was experienced in shielding thermometers used in taking the temperatures in the furnace bonnets against radiant heat.

The Test Results

It is recognized that the measurement of heat in an air stream by thermometers hung in that stream and the measurement of air flow by an anemometer is an inexact method of measuring heat quantities and is subject to considerable error. The data obtained, therefore, are not submitted as exact information, but as an approximation of a furnace field test after installation. The furnace was operated for

one hour before the official test commenced in order to bring furnace, ducts and rooms up to constant temperature. Readings were taken every 15 min. throughout the three-hour test and averages used in making the calculations.

Data of test for 3-hour period from 9:30 a. m. until 12:30 p. m.:

Average temperature of incoming air....25.3 F
Air delivery as measured in fresh air shaft....17400 c.f.m.
Average temperature air discharged from furnace....88.2 F
Average temperature rise through furnace....62.9 F
Barometer....29.96
Gals. oil burned in 3-hour period....33.37
Average room temperature throughout test....68 F
Relative humidity outdoors....95%
Relative humidity in rooms....28%
Wind—S.W....15 m.p.h.
Temperature of gas in breeching....298 F
Draft in stack....0.45 in.
Draft in furnace....0.14 in.

Calculations from Test:

Heat absorbed by air in 3 hours, $17400 \times 0.24 \times 0.075 \times 62.9 \times 180$ ....3,546,050 Btu
Heat in burning oil in 3 hours, $33.37 \times 7.868 \times 18{,}891$ ....4,959,929 Btu

Overall furnace efficiency.... $\frac{3{,}546{,}050}{4{,}959{,}929} = 71.5\%$

To compare the heat absorbed by the air and the total heat required to take care of all heat losses in both heating and ventilating, the transmission and infiltration losses for the building are presented as follows:

Walls....U = 0.216
Basement floor....U = 0.556
Doors....U = 0.382
Windows....U = 1.13
Plaster ceiling....U = 0.551

Transmission Losses:

N & E glass transmission, $982 \times 1.13 \times 42$.... 46,600
S & W glass transmission, $900 \times 1.13 \times 42 \times 1.15$.... 49,120
N & E wall transmission, $7577 \times 0.216 \times 42$.... 68,740
S & W wall transmission, $7756 \times 0.216 \times 42 \times 1.15$.... 80,915
Doors transmission, $194 \times 0.382 \times 42$.... 3,113
2nd floor ceiling transmission, $4875 \times 0.551 \times 10$.... 26,860
Basement floor transmission, $4875 \times 0.556 \times 15$.... 40,657

Total transmission losses per hour....316,000 Btu

Infiltration losses:

Doors.... 14,000
Windows.... 29,500

Grand total per hour....359,500 Btu

or 1,078,500 for the three-hour period.

$\frac{(88.2 - 68)}{62.9}(3{,}546{,}000) = 1{,}139{,}000$ Btu surplus heat supplied by the air.

The difference between 1,139,000 and 1,078,500 may be accounted for in
(a) Heat absorption by the building materials.
(b) Errors in allowance for infiltration.
(c) Errors in measurement of air volume and temperatures.

Analysis of the fuel oil used throughout the test:

Gravity Baumé....18-20°
Weight per gallon....7.868 lb.
Sulphur....0.54%
Btu per lb....18,891

A flue gas analysis which was made to determine the stack losses resulted as follows:

Carbon Dioxide.... 5.8%
Oxygen....13.5%
Carbon Monoxide.... 0
Nitrogen....80.7%

Calculations from this analysis determine that there was 178 per cent excess air and stack losses of 15.78 per cent. The large amount of oxygen and the great excess air is due to the strong stack draft which would have been required in burning coal but was not required in burning oil. This indicated the necessity of installation of a check draft.

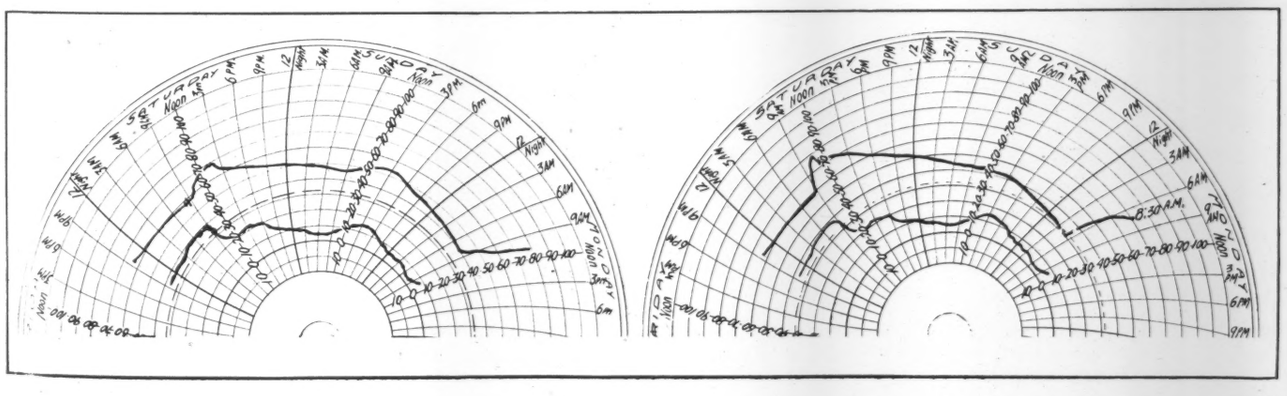
Importance of Zoning Control

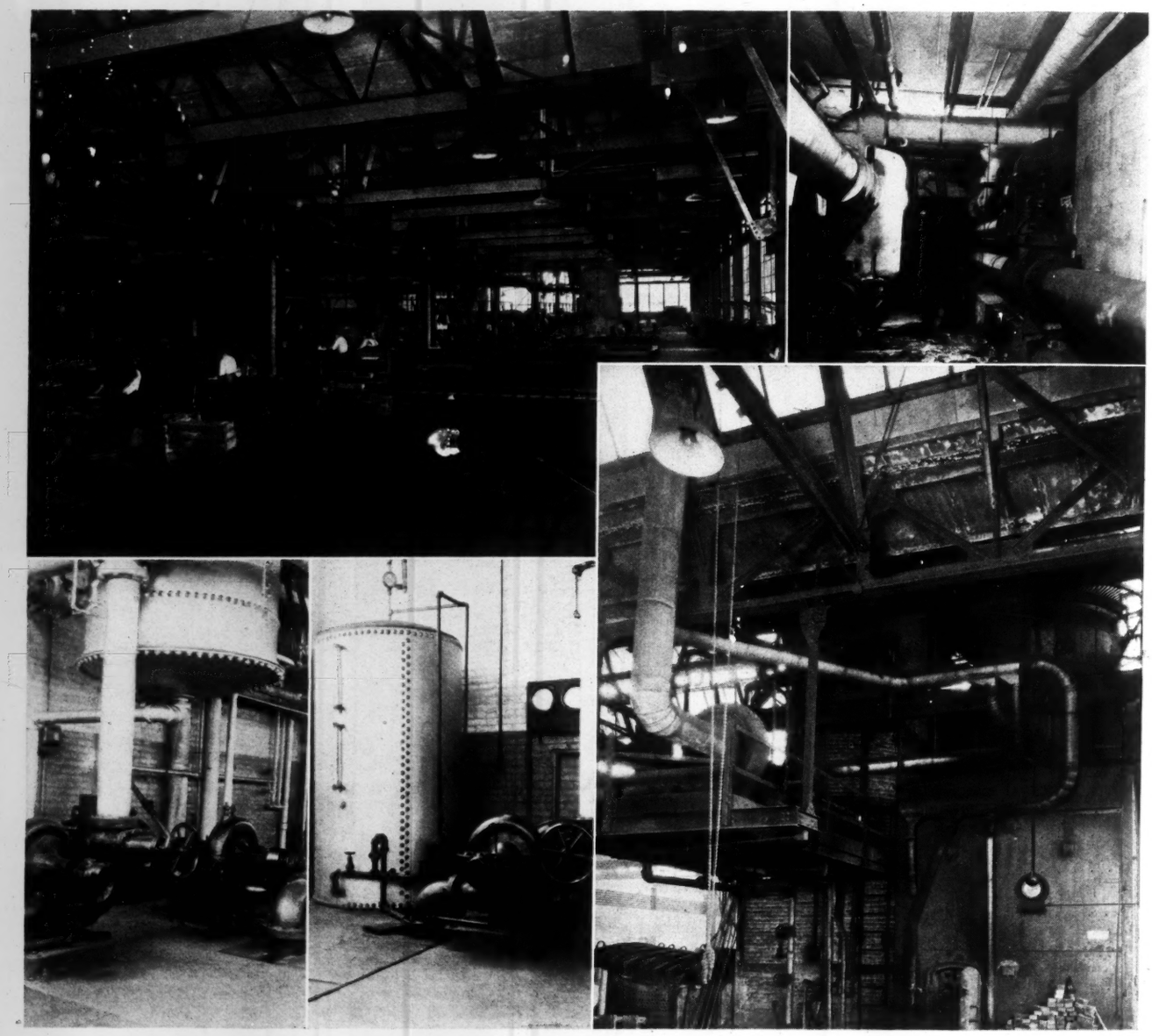
The effects of solar heat are receiving considerable attention these days. What solar heat may do to the temperature of a classroom is indicated by the two temperature charts in Fig. 3, which show the outdoor temperature curve over Sunday with corresponding indoor temperature curves. The rise in temperature in a south room on Sunday afternoon over that in a north room is marked. The furnace was shut down from 2:00 p. m. Saturday until 7:00 a. m. Monday. Engineers designing heating and ventilating systems are becoming convinced of the desirability of separating the control of the south and north rooms in order to take care of the effects of solar heat. Satisfactory regulation without such separation of the system has proved difficult.

FUEL OIL CONSUMPTION RECORD
Month—February 16-March 16—1932

DATE	Average Outdoor Temperature Between 8:00 A.M. & 3:00 P.M. in Degrees Below 70	WIND		Pounds Oil Burned	Pounds Oil Burned Per Degree Outside Temp. Below 70 Degrees
		Direction	Velocity M. P. H.		
Feb. 16	34	S. E.	18	640	18.8
" 17	36	W.	30	640	17.8
" 18	44	S. W.	22	750	17.0
" 19	40	N. W.	30	630	15.8
Feb. 23	54	N. W.	22	825	15.2
" 24	40	N. E.	12	680	17.0
" 25	25	S. W.	12	565	22.6
" 26	22	S. W.	10	400	18.2
Feb. 29	29	N. W.	21	600	20.6
Mar. 1	39	S. E.	12	645	16.5
" 2	37	N. E.	14	610	16.5
" 3	36	N. E.	12	590	16.4
" 4	34	N. E.	21	585	17.2
Mar. 7	65	N. W.	27	1165	17.9
" 8	66	N. W.	21	1000	15.1
" 9	59	N. W.	14	900	15.2
" 10	60	N. W.	12	840	14.0
" 11	53	N. W.	18	760	14.4
Mar. 14	56	N. W.	12	930	16.6
" 15	44	S. E.	11	690	15.6
" 16	31	N. W.	12	550	17.7
				AVERAGE	17.0

Fig. 3—Left—Indoor temperature curve for room with south exposure. Right—Curve for room with north exposure. These charts indicate desirability of separating control of north and south rooms to take care of effects of solar heat





Cuts Heating and Compressed-Air Cost with Waste-Heat Boilers

By Thomas R. Shaver

IN many types of industrial plants, large amounts of waste heat are available in connection with the process carried on which can be utilized to advantage by one means or another for space heating. Waste-heat boilers find applications in many such plants, and in several types of industries their application should be carefully investigated when economical plant heating is being considered.

The malleable industry has a large number of plants operating waste-heat boilers. The paper industry also offers opportunities for heat-saving equipment of this type, where the steam generated is usually used in turbines and is exhausted for heating. In the cement industry waste-heat boilers are used, but the use of steam so generated for heating is comparatively small and is usually confined to heating the offices. Another field developing at the present time is the use of water heaters and waste-heat boilers in connection with large diesel engines, the steam being used for heating or piped to process work.

The heating plant at the Belle City Malleable Iron Co. plant, Racine, Wis., is a good example of what

Opposite (upper left)—Malleable foundry, showing arrangement of heating coils. Upper right—View in pipe tunnel, showing hot-water, steam, boiler feed and compressed-air lines. Lower right—One of the waste-heat boilers. Lower center—Expansion tank and circulating pump. Lower left—Water heater and circulating pumps

may be accomplished along these lines. A forced-circulation hot-water system furnished with exhaust steam from steam-driven air compressors is used. Steam is generated in two waste-heat boilers, both equipped to operate as direct-fired boilers at times when no waste heat is available. The steam heats the water which is circulated.

The system was designed to produce the required inside temperatures in the various buildings with an outside temperature of —10 F. Temperatures used as the basis of the design are: Steel foundry, 55; malleable foundry, 60; core room, 70; shipping building, 55; maintenance building, 65; cleaning room, 55; and garage, 55 F.

The sand bins have sufficient radiation to prevent freezing of the moisture in the sand. The general office building is heated by a separate small boiler.

Steam from Waste Heat Drives Air Compressors; Exhaust Heats Water

The fact that more than enough steam to furnish the compressed air, an important form of power in a foundry, and all the heat required for the buildings during normal working hours, is otherwise wasted out of the malleable foundry melting furnace stacks, led to the use of the waste-heat boilers. Fig. 1 shows their location, one at each of the two 20-ton melting furnaces. These furnaces are fired with pulverized coal, each being equipped with a unit pulverizer driven by a 40-hp. motor and with a blower driven by a 15-hp. motor. Furnaces are so arranged that exhaust gases can be passed up the furnace stack or passed through the waste-heat boilers and then exhausted through a short boiler stack equipped with a slow-speed exhauster driven by a 15-hp. motor, for induced draft. Exhaust gases leave the melting furnaces at maximum temperatures of 2600 to 2800 F. Boilers are of the vertical water-tube type, rated at 450 hp. and generating steam at 200 lb.

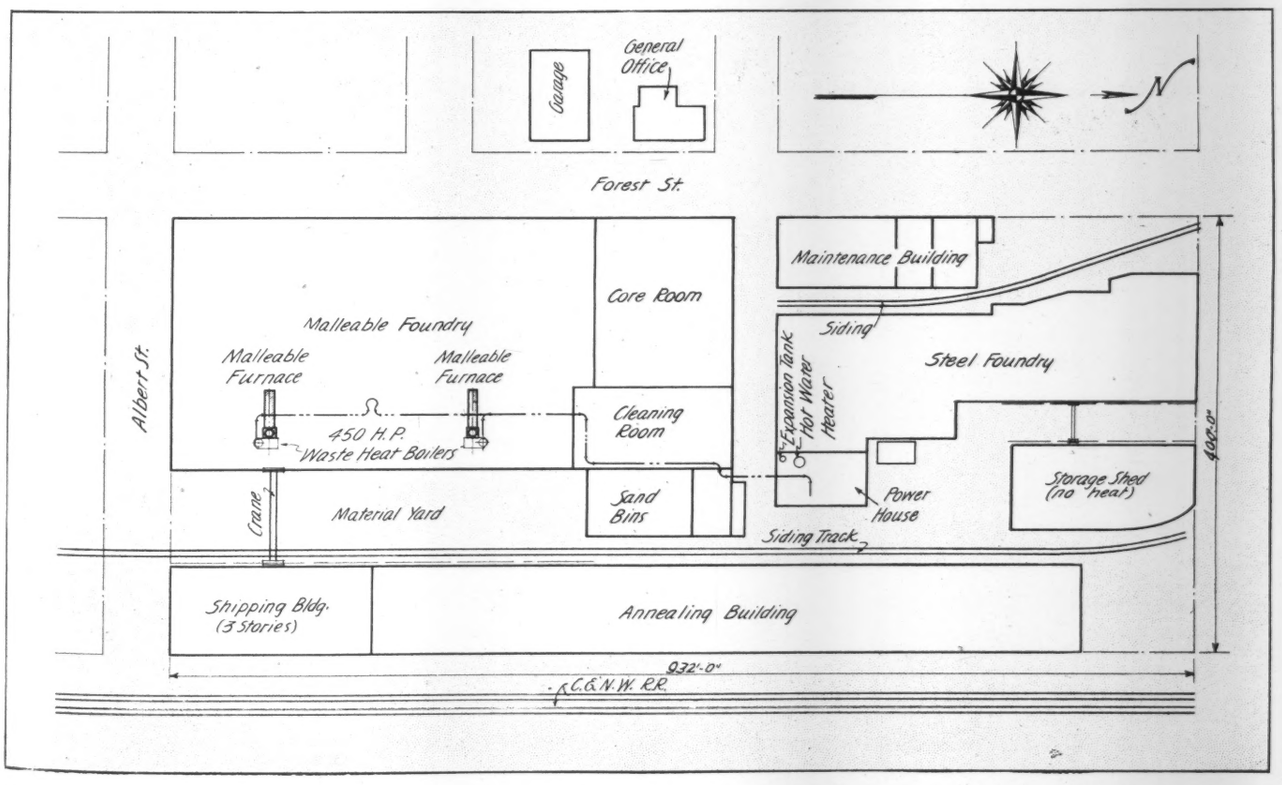
One Boiler Is Direct Fired at Night

During normal production in the foundry the melting furnaces are in operation from about 5:00 a. m. —when fires are started—until about 5:00 p. m., when the last iron is poured. During this time the exhaust furnace gases are used to fire the two boilers, and steam is generated to operate equipment in the power house. Both boilers are equipped with an auxiliary pulverizer and one of the boilers is fired direct during the night and on Sundays and holidays.

Utilization of the Steam

The steam is passed through an 8-in. overhead main through the malleable foundry and cleaning

Plan of the Belle City Malleable Iron Co. plant showing the location of the two waste-heat boilers which generate steam for driving the air compressors and heating the water for the forced-circulation hot-water heating system. Steam is also furnished to the turbine driving the circulating water pump and to the boiler feed pumps



room to a point approximately 40 ft. south of the power house, where the main drops down and into a pipe tunnel under the roadway just south of the power house. This tunnel is the only one in the plant, and carries the hot-water heating mains, and boiler feed and compressed air piping in addition to the steam main.

The principal use made of the live steam is in driving two air compressors. Both compressors are of the reciprocating type, one of 300 hp. with a capacity of 2000 c.f.m., and the other 100 hp. with a 700-c.f.m. capacity. Steam is also used for driving a centrifugal-type turbine-driven circulating water pump of 1200-g.p.m. capacity, and two cross-connected boiler-feed water pumps.

The exhaust steam from the high-pressure equipment in the power house is passed through the circulating water heater and feed-water heater where practically all is condensed. The hot condensation from the heaters is delivered by gravity to a small centrifugal pump which in turn discharges it into the feed-water heater. The feed-water heater has a float-controlled city water make-up line to maintain a fixed level in the heater. The boiler feed pumps are controlled automatically from the boilers by feed-water regulators which operate in conjunction with governors installed in steam lines to pumps.

The circulating water heater is a vertical steam-tube type. Steam enters at 5 lb. pressure, and water can be heated to 210 F by exhaust steam alone. When steam-driven compressors are not running (at night and on Sundays and holidays), steam is passed through a pressure-reducing valve before entering the heater. The drop in water temperature in the circulating system is normally about 30 F.

To provide for times when steam is not available there are two motor-driven air compressors, and one motor-driven pump for circulating heating water for emergency use. Each boiler has the usual accessories, including an injector and automatic stop and check valves at the steam outlet, blow-off valve and cock, high and low alarm water column, and steam gage.

Long runs are the rule with the heating coils, the longest being a 17-pipe coil 360 ft. long without a break along the west wall of the malleable foundry building. In other cases the coils are cut where doors interfere with continuous runs. In general, coils are run from the supply end (nearest the power house) along the walls to the opposite end of the building, then are collected together and the water is returned by one or more overhead coils. Water expansion is provided for by a closed tank in the power house.

All piping is standard-weight wrought steel. The steam line has extra-heavy van stone flanged joints and extra-heavy flanged fittings. All other fittings are standard threaded, except the boiler feed line, which has extra-heavy threaded fittings. All water pipes are exposed within the buildings except the short lengths in the one tunnel, and underground pipes in tile serving the maintenance building, shipping and annealing buildings, and garage. Expansion in steam and water piping is taken by pipe bends.

A. A. Wickland & Co., Inc., were the engineers in charge of the design and construction of the system.

Heat Loss from Underground Steam Lines Tested

DURING the summer of 1929, there was installed in Indianapolis approximately thirteen miles of new underground steam-heating mains; since these mains were installed at one time and with the same method of insulation, it afforded an excellent opportunity for determining the losses from such mains.

These mains, with one feeder line from the plant, form a network within an area approximately one mile long and one mile wide. Insulation consists of 2-in. pipe covering on 10-in. and larger pipe and 1½-in. thickness on 8-in. and smaller pipe within a conduit structure. Soil conditions are mostly gravel, and the depth of mains varies from 3 to 6 ft. coverage.

To obtain data on the heat losses from this distribution system, a test was carried out from Feb. 21 to March 21, 1930. For this test the valves between this distribution system and another connected system were closed and all consumers' meters were read on the first and last days of this test period.

The actual steam loss and unaccounted for, considered here as radiation losses from the underground steam system, was 5,902,700 lb. for the period of 28 days, which equals an average of 8,784 lb. per hr. or approximately 8,313,180 Btu. (Heat of vaporization of steam at 10 lb. gage considered as 946.4 Btu per lb.)

Considering the temperature of steam at the above pressure to be 240 F and the temperature of the ground at the depth of the steam mains as 50 F results in a temperature difference of 190 F. Reliable heat loss data for this temperature difference show the heat loss from 8 in. diameter bare pipe in still air to equal 1,125 Btu per hr. per ft. 68,400 lineal feet × 1,125 equals 77,017,500 Btu loss per hr. total for this amount of bare pipe in still air.

For actual heat loss compared to heat loss from bare pipe, it shows:

$$\frac{\text{8,313,180 Btu actual loss per hr.}}{\text{77,017,500 Btu loss per hr.—bare pipe}} = 10.8\%$$

This shows that the actual heat loss was 10.8 per cent compared to the heat loss from an equivalent amount of bare pipe in still air, an efficiency of 89.2 per cent for the pipe line insulation.

This insulation efficiency is obtained from the test on the entire underground distribution system including manhole losses and compares very favorably with the efficiency of the pipe covering used, which is quoted at 90.7 per cent, inclusive of manhole losses and other extraordinary conditions which affect the heat losses of the system as a whole.

The losses measured amounted to 23.4 per cent of the total steam delivered to the system. The system at the time of the test was loaded to about 20 per cent of the capacity and available load. Results indicate that under fully loaded conditions the losses would amount to about 5 per cent of the steam delivered to the mains.

—From committee report presented at 1932 meeting of *National District Heating Association*.

Changes Pipe Size, Nets 31% on Cost

• • • • correctly sized suction line on ammonia compressor cuts power charge

By Herman Vetter*

PICKING the proper size of pipe to use in any piping system depends on a consideration of the allowable pressure drop and the cost of power necessary to overcome the friction of the line. In a refrigeration plant, the effect of the pipe size, vapor velocity, and the cost of power frequently is not given the attention warranted its importance. This article points out the method of figuring the effect the size of the low-pressure ammonia or suction lines has on the cost of the power consumed.

Calculating Volume of Vapor Handled

To plot velocity curves the volume of vapor to be handled per minute for various conditions must first be calculated. To simplify calculations a fixed condensing pressure is assumed.

The volume of vapor depends on the quantity of liquid ammonia to be evaporated per minute to produce one ton of refrigeration. Assuming a condensing pressure of 166 lb. gage and an evaporator pressure of 5 lb. gage, standard tables on the properties of ammonia give:

Heat content of liquid at 166 lb.....143.5 Btu/lb.
Heat content of liquid at 5 lb..... 24.4 Btu/lb.
Latent heat at 5 lb. pressure.....581.6 Btu/lb.
Volume of vapor at 5 lb. pressure.... 13.7 cu.ft./lb.

In a *steam plant* the feed-water to the boiler is heated to the temperature in the boiler. In a *refrigeration plant* the liquid ammonia fed to the evaporator must be cooled from the condensing temperature to the evaporator temperature. This cooling is done by evaporating a portion of the liquid; consequently we must determine this amount before we can find the total amount of vapor to be handled.

$$143.5 - 24.4 = 119.1 \text{ Btu/lb.}$$

Therefore, the amount of heat to be extracted to cool the liquid ammonia from 166 lb. condensing pressure and 90 F temperature to 5 lb. pressure and —17.2 F equals 119.1 Btu for each pound of ammonia circulated.

The total effective refrigerating effect of evaporating one pound of ammonia under the above conditions is therefore:

Latent heat at 5 lb 581.6
To cool from 90 to —17.2 F.119.1

462.5 Btu/lb.

One ton of refrigeration = 200 Btu/min. Therefore:

$$\frac{200}{462.5} = 0.4324 \text{ lb./min./ton refrigeration.}$$

The volume of vapor to be handled per minute per ton of refrigeration is therefore:

$$0.4324 \times 13.7 \text{ cu. ft.} = 5.92388 \text{ cu. ft./min.}$$

In like manner the volume of vapor is determined for

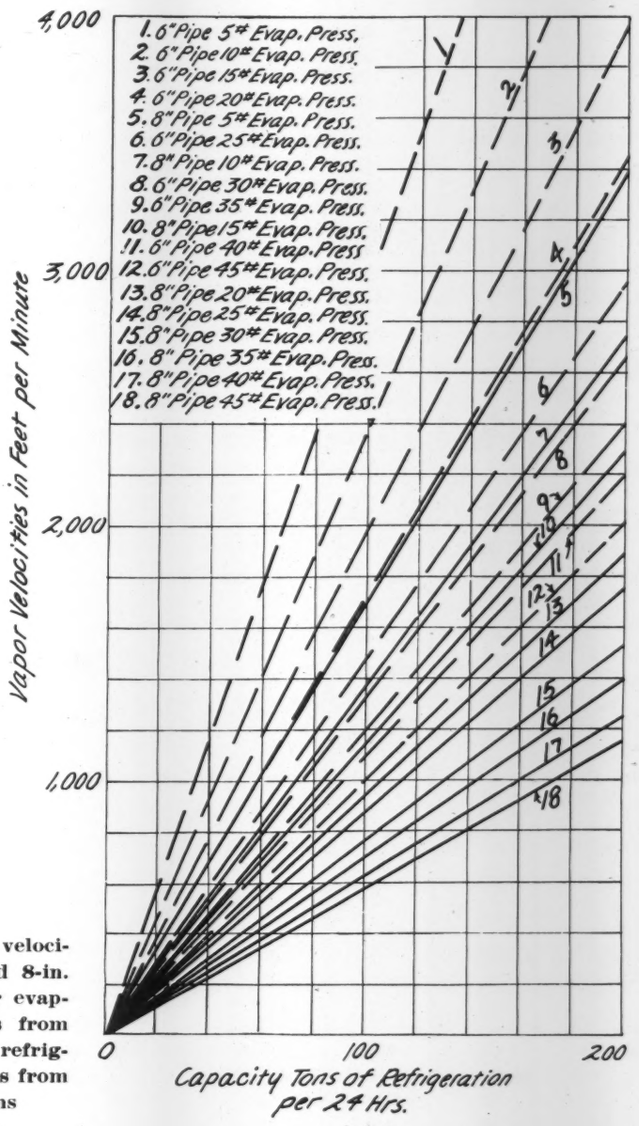

1. 6" Pipe 5# Evap. Press.
2. 6" Pipe 10# Evap. Press.
3. 6" Pipe 15# Evap. Press.
4. 6" Pipe 20# Evap. Press.
5. 8" Pipe 5# Evap. Press.
6. 6" Pipe 25# Evap. Press.
7. 8" Pipe 10# Evap. Press.
8. 6" Pipe 30# Evap. Press.
9. 6" Pipe 35# Evap. Press.
10. 8" Pipe 15# Evap. Press.
11. 6" Pipe 40# Evap. Press
12. 6" Pipe 45# Evap. Press.
13. 8" Pipe 20# Evap. Press.
14. 8" Pipe 25# Evap. Press.
15. 8" Pipe 30# Evap. Press.
16. 8" Pipe 35# Evap. Press.
17. 8" Pipe 40# Evap. Press.
18. 8" Pipe 45# Evap. Press.
4,000
3,000
2,000
1,000
0
100
200
Vapor Velocities in Feet per Minute
Capacity Tons of Refrigeration per 24 Hrs.

Fig. 1—Vapor velocities in 6-in. and 8-in. suction lines for evaporator pressures from 5 to 45 lb. and refrigerating capacities from 0 to 200 tons

*Consultant on refrigerating engineering, Los Angeles, Calif. Member *A. S. R. E.*

other evaporator pressure. These quantities have been determined as follows:

Evaporator Pressure, Lb.	Cubic Feet of Vapor per Min.	Theoretical Horse Power per Ton Refrig.
5	5.9238	1.52
10	4.7686	1.35
15	3.9834	1.19
20	3.4295	1.05
25	2.9812	0.96
30	2.6829	0.88
35	2.4177	0.81
40	2.2060	0.73
45	2.0271	0.66

Determining the Horsepower Required

The theoretical horsepower for the above conditions is determined as follows:

Heat content of superheated ammonia at 166 lb. and 280 F equals 753.9
Heat content of ammonia at 5 lb. and —17.2 F equals 606.2

147.7 Btu/lb.

147.7 Btu × 0.4324 lb. = 64.8754 Btu/min./ton refrigeration.
1 Btu/min = 42.44 hp.

$$\frac{64.8754}{42.44} = 1.52 \text{ theoretical hp. per ton of refrigeration.}$$

The theoretical horsepower for the other evaporator pressures were determined in like manner.

The velocities for the different evaporator pressures and for different size pipe lines can now be determined. Fig. 1 illustrates the velocities for 6-in. and 8-in. pipe lines for evaporator pressures from 5 lb. to 45 lb. inclusive and for refrigerating capacities up to 200 tons per 24 hours.

The pressure drop for pipe lines from 2-in. to 8-in. is shown in Fig. 2. The formula used is:

$$p = \frac{L v^2}{25{,}000\, d}$$

p = loss of pressure, in oz. per sq. in.
d = inside diameter of pipe, in in.
L = length of pipe, in ft.
v = velocity in ft. per sec.

The increase in power due to the pressure drop in the suction line for the different evaporator pressures is:

Evaporator Pressure, Lb.	Increase in Horsepower Per Oz. Drop in Pressure per 100 Ft.
5	0.0116
10	0.0106
15	0.0100
20	0.0093
25	0.0056
30	0.0049
35	0.0047
40	0.0046
45	0.0046

Example Indicates Possible Power Saving

The following example indicates how an increase in power due to friction in the piping will increase the power cost.

Capacity of plant	= 100 tons refrigeration.
Size of suction line	= 6 in.
Evaporator pressure	= 15 lb.
Velocity in suction line	= 1980 ft./min.
Pressure drop at compressor	= 0.73 oz.

The saving effected if an 8-in. instead of a 6-in. suction line were used would be:

Velocity in 8-in. line = 1150 ft./min.
Pressure drop = 0.16 oz.

The difference in pressure drop is therefore 0.73 — 0.16 = 0.57 oz.

0.57 oz. × 0.01 hp. × 100 tons = 0.57 hp. per hr., or 410 hp. per month or 306 kw. per month.

In the case considered the difference in cost of installing a 6-in. or an 8-in. line was determined to be $116. Taking advantage of the saving in power that would be made if the larger line were installed, the computations indicate that a return of 31 per cent would be made on this added investment. The equivalent length of the suction piping is assumed as 100 ft.

Note: The thermodynamic properties of ammonia used in this article are from Circular No. 142 of the Bureau of Standards.

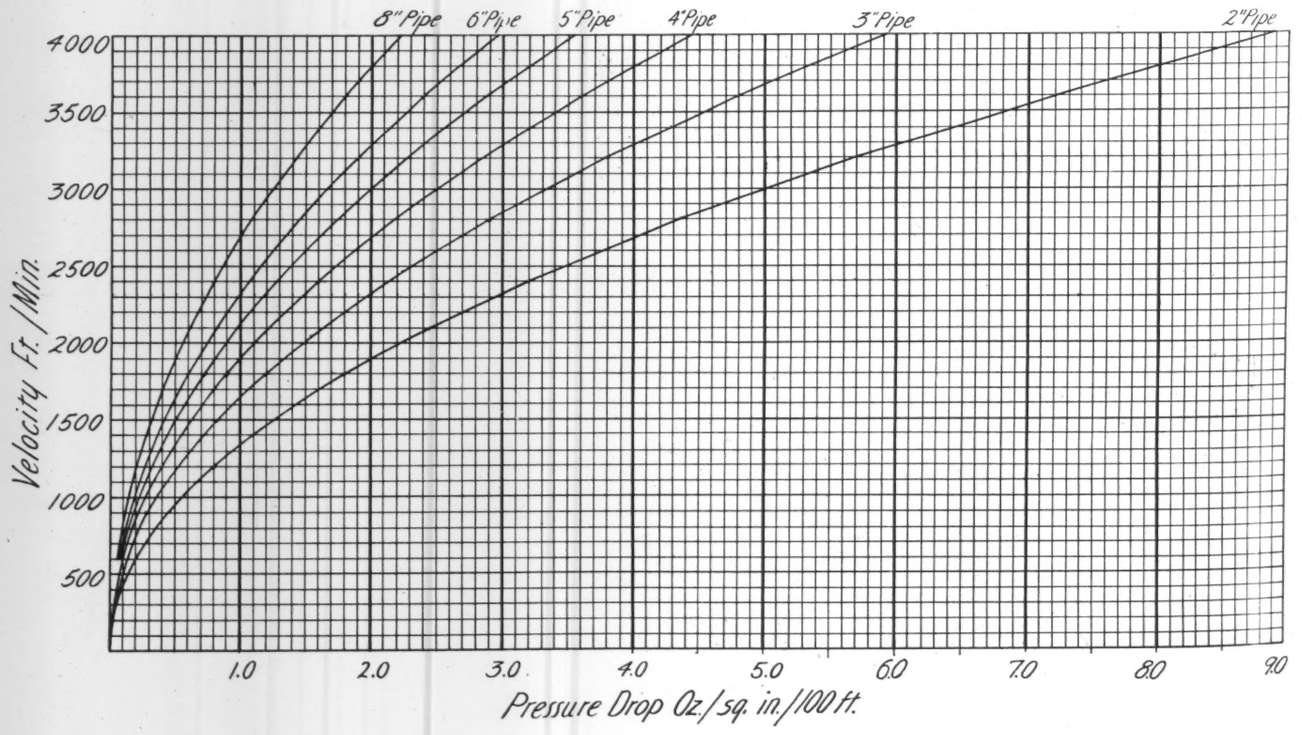


Fig. 2—Pressure drop in various sized piping for different velocities

Plants Pay for Proper Piping, Whether They Have It or Not . . . . . .

. . . R. C. Doremus* discusses Vetter article

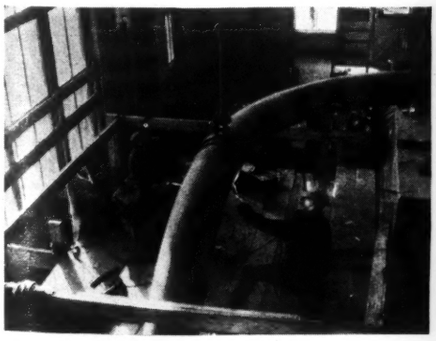
PROPER design of piping in a refrigerating plant is as important a factor in economical operation as the choice of well-balanced equipment. Either one without the other leaves much to be desired. Very frequently refrigerating plants of various types and sizes will be found to comprise equipment of satisfactory capacity for a desired production and yet the items of apparatus are connected with piping utterly inadequate for the load. Rarely is the piping ever found to be so large as to be open to criticism on this point. Frequently little thought is given to the velocities of vapor involved, whereas the pipe size for a given tonnage at low back pressure must be considerably larger than the pipe size for the same tonnage handled at higher back pressure. Later, additions are made which were not originally planned on, more equipment installed, and all "connected" to the same piping until, like Topsy, it looks as if it "just grew." Such developments lacking in engineering and foresight spell foolish economy.

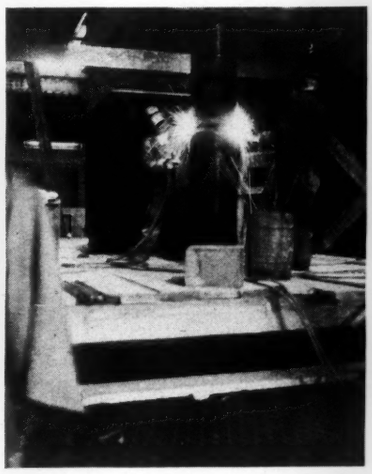
In any refrigerating plant, piping is used in more or less liberal quantities, as so much surface for either the evaporating or condensing sides of the system. Other piping is used between items of equipment as the conduit for the fluids of the system, whether they be liquid or vapor, and this piping must serve satisfactorily just as the major items of equipment. In the case of liquids, such as liquid ammonia, water, brine, etc., the engineer may be quite familiar with the frictional losses and proper capacities. On the other hand, the vapor velocities are frequently slighted or overlooked. Mr. Vetter's article covers the methor of calculating vapor velocities for ammonia very well; a study of it will enable any engineer to check over his own conditions to determine for himself whether or not suction pipe sizes are ample. If the lines are far too small, production is bound to suffer. The equipment may be suitable in capacity for 100 tons of refrigeration and yet a small suction line will restrict the flow of vapor, on which capacity depends, to a much smaller amount—say 80 to 90 tons—due to the bottle-neck in the system.

Such instances can be subsequently remedied, and many have been, by changes in piping to shorter runs of adequate sizes to meet conditions. The owners of many of the larger plants where small piping was installed are cognizant of such costs and possible savings, and are making piping changes to affect lower costs in operation. At the same time, many of the smaller plants, where most of the indiscretions in pipe sizes occur, continue to operate under this handicap either through a lack of realization of their condition or through endeavoring to operate on a minimum budget for capital expenditures of this nature.

If the lines are just slightly small, production capacities of the plant may be obtained by forcing the fluids through the lines at abnormally high velocities, but at the expense of excessive friction pressure drops which means high power bills. It will be seen that it is essential to have an ammonia suction line of proper size for economical velocities and, of course, it is preferable to make these lines as short and direct as possible from evaporator to compressor. If piping of suitable size is not paid for as a capital investment, it will have to be paid for, several times over perhaps, in power bills—the only difference is that the bookkeeper will enter the expense as "operating" in the ledger.

* George B. Bright Co., Refrigerating Engineers and Architects, Detroit, Mich.



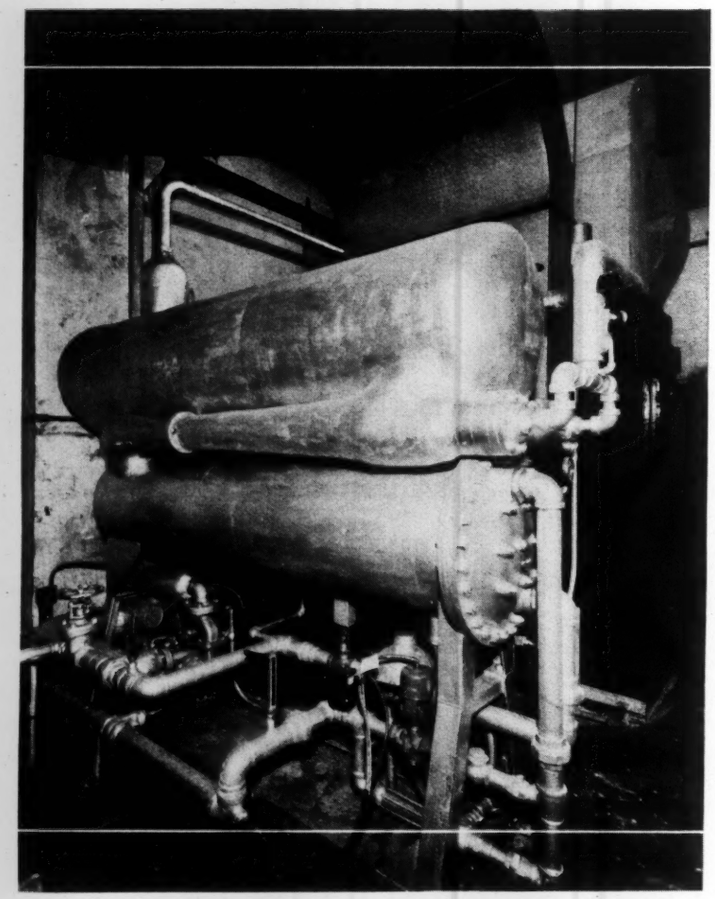


Left—Making a position weld. Center—Welding in process. Right—Oil-fired furnace for annealing joints in place

Illustrations from "Pipe Welding for a High-Pressure Steam Power Plant" presented at annual meeting of American Welding Society by E. B. Severs and W. P. Gavit, April, 1932.

Installs Cooling System June 1st; Summer's Business Increases 60%

Steam-ejector cooling system supplies conditioned air to restaurant . . . 100-lb. steam bought from adjoining building . . . 27-ton capacity



The steam-ejector water cooler which furnishes water ranging from 35 to 42 F for cooling air supplied to Maurice's restaurant, Chicago. Temperature of the water is controlled by the amount of vacuum produced by the ejector, which is supplied with 100-lb. steam purchased from an adjoining building

BUSINESS men are pretty well in agreement with the idea that it is wise to take advantage of present conditions and new money-saving devices by rehabilitating industrial plants and other larger structures now to put them in readiness for the business improvement on the way. Such a movement, if undertaken earnestly and wisely by American business, will be of benefit not only to the seller and the buyer but to general conditions, it is believed.

An example of the application of these principles may be seen in the cooling system in Maurice's, one of the leading restaurants of Chicago, which went into operation on June 1 of this year.

Here, cost was a factor. Due to new engineering developments the complete cooling system cost installed much less than a cooling system in a nearby restaurant cost three years ago.

Profit on the investment was a factor. Business done by this restaurant increased this year sixty per cent over that of any previous summer.

Preparation for the "return of prosperity" was a factor. The management believed that the time was ripe to get ready; their increase in business showed that their judgment was good.

System Uses Steam Ejector to Produce Vacuum

The design of this cooling system is based on the steam ejector principle. The steam ejector as applied to cooling of railway passenger cars was described in the April, 1932 issue of HEATING, PIPING AND AIR CONDITIONING. Several cooling systems based on this principle have been developed. In all, the ejector is used to induce a

vacuum in a water container. Under the high vacuum that may be obtained this way, water may be cooled almost to the freezing point. The water is then circulated through coils over which the air to be cooled is blown. The temperature of the water is controlled by the amount of vacuum, and ranges from 35 to 42 F. The steam pressure used is 100 lb.

Cooling of Restaurants

In restaurants without mechanical cooling equipment, often the differential between the temperature of the air inside and outside is 20 F; when outside air is 70 F the air inside the restaurant may be 90 F due to the heat from patrons, lights and cooking. Fans are commonly used for cooling restaurants. They serve a very useful purpose in ventilating, removing odors and effecting a degree of comfort but restaurant owners often are disappointed in trying to cool a restaurant by drawing 95 F outside air into a restaurant that is already uncomfortably warm.

Conditions Maintained by System

Maurice's restaurant has a capacity of 50,000 cu. ft.; the floor area is 4,000 sq. ft. It is enclosed on all sides by other buildings. In the design of the cooling system these factors and the heat from patrons, lights and cooking, were carefully evaluated.

Design of the system was based on an occupancy of 250 people, each giving off 240 Btu per hr. of sensible and latent heat and to provide comfortable temperature and humidity conditions at all times during the summer months. For instance, at 90 to 100 F with high relative humidity outside, the restaurant is maintained at 80 F and 50 per cent relative humidity. The aim is to maintain ten to fifteen degrees difference between the inside and outside temperature during warm weather. When the temperature is 85 outside, the inside temperature is maintained at approximately 75 F. With an 80 F outside temperature, the inside temperature is at 74 F. The equipment usually is in operation during the summer on week days from 11:30 a.m. to 4:15 p.m., and 8:00 to 10:00 in the evening; on Sundays 11:30 a.m. to 5:00 p.m.

Thermostat Starts Cooling System When Outside Temperature is 60 F

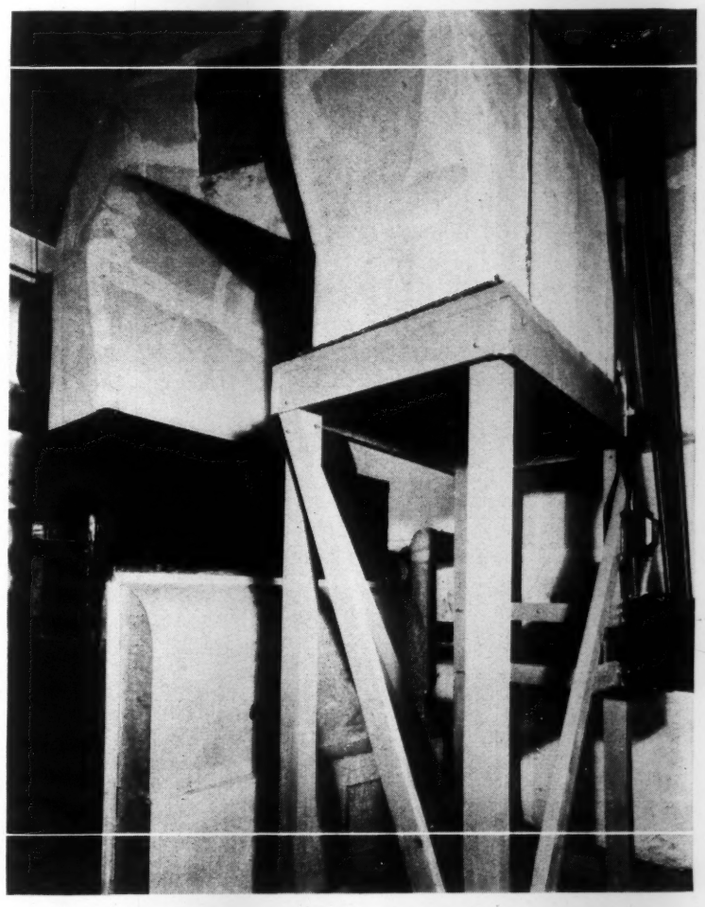
The heat input from patrons, lights and cooking is such that the thermostat starts the cooling system when the temperature outside rises to 60 F. At this outside temperature comfortable inside conditions are maintained by introducing air into the restaurant at 50 F. This relationship is based on the known rate of heat loss of the building and the known amount of heat generated in it. A restaurant differently situated and with a different heat input and heat loss would have a different differential between inside and outside air and would require that air be introduced at a different temperature. It is the experience and skill in estimating this differential upon which the efficiency of the system depends.

Steam, Water and Power Required

The cooling equipment consists of a steam ejector water cooler, the piping through which the cooled water circulates, fans which draw air around the pipe coils and duct work which delivers 8000 c.f.m. of cooled air to the restaurant. This system is under complete automatic control. It has a capacity of 27 tons of refrigeration and requires 950 lb. of steam per hr. on full load.

The steam is purchased at $1.00 per 1,000 lb. from an adjoining building. The cost of steam this summer was: June, $81.00; July, $199.00; August, $180.00; September, $177.00.

Approximately one-fourth of this steam load was used for cooking and the remainder for cooling. Condensing water was required at the rate of 180 g.p.m. during the operation of the equipment. One 5-hp. and one 3-hp. motor are a necessary part of the air-conditioning equipment.



The fan and enclosed air coolers installed in Maurice's restaurant. Eight thousand c.f.m. air, cooled to the proper temperature to maintain comfortable conditions, is delivered to the restaurant. Heat gain in the building is such that the cooling equipment goes into operation when the outdoor temperature reaches 60 F

Recent Trade Literature

Circulators: William J. Lohman, Inc., 92 Warren St., New York City; folder describing portable air circulators for use in meat-packing plants, etc., for preventing ceiling condensation and securing uniform temperature and humidity.

Drives: The Rockwood Manufacturing Co., Indianapolis, Ind.; folders describing function, installation methods, specifications, etc., of pivoted motor bases for short-center flat belt drives; adjustable arms make them applicable to all types and weights of motors.

Fittings: Taylor Forge & Pipe Works, P. O. Box 485, Chicago, Ill.; eight-page bulletin discussing the welding of piping and the application of forge seamless steel fittings.

Fittings: Parker Appliance Company, 10320 Berea Rd., Cleveland, Ohio; 40-page catalog giving complete information on fittings for copper tubing including engineering and installation data and general information.

Mixers: The Patterson Foundry & Machine Co., East Liverpool, Ohio; four-page bulletin describing portable motor-driven mixers for mixing, blending or agitating paint, oil, solvents, chemicals, dye, varnish, ink, tonics, sizing, and similar uses. The device is mounted on the edge of the tank.

Pipe: Republic Steel Corporation, Republic Bldg., Youngstown, Ohio; price card on copper molybdenum iron pipe, embodying simplified method of computing prices.

Pipe Bending: A. M. Byers Company, Clark Bldg., Pittsburgh, Pa.; 12-page treatise on the principles and practices of bending wrought-iron pipe, including a discussion of the fundamentals of bending pipe.

Pumps: Columbus Steam Pump Works Company, Columbus, Ohio; 24-page bulletin describing single cylinder, double acting and duplex steam pumps for general service, boiler feed, piston pump tank and low vacuum service, etc.

Pumps: Kingsford Foundry & Machine Works, Oswego, N. Y.; 12-page bulletin on single stage, double suction centrifugal pumps and 12-page bulletin on centrifugal pumps for pulp and paper plants. Also 24-page catalog devoted to engine-driven pumps and engines.

Soot Blowers: Diamond Power Specialty Corporation, Detroit, Mich.; four-page folder describing automatic soot blower systems for cleaning fire-tube and water-tube boilers.

Stokers: Detroit Stoker Company, General Motors Bldg., Detroit, Mich.; 16-page bulletin describing in detail the features of an underfeed stoker applicable to new and existing boiler installations. Performance records of typical installations are included.

Ventilators: Truscon Steel Company, Youngstown, Ohio; six-page folder on non-mechanical window ventilator for office buildings, etc., designed to muffle noise.

Welders: Universal Power Corporation, 12367 Euclid Ave., Cleveland, Ohio; two 4-page bulletins describing motor-generator portable and stationary arc welders with shunt inductor to give smooth soft arc; a portable gasoline-electric road trailer model is also described.

Welders: Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa.; data sheet on 100 ampere a-c. welder with short wave arc control, weighing 400 lb.

Welders: The Imperial Electric Company, Akron, Ohio; data sheet on a-c. or d-c. motor-driven portable arc welder.

Welding Rod: The Linde Air Products Company, 205 E. 42nd St., New York City; 20-page booklet giving physical and welding characteristics of bronze welding rod with which tensile strengths in joining steel of 56,000 to 60,000 lb. per sq. in. have been obtained, with ductility over 30 per cent. Recommendations for proper welding are included and applications are described.

Conventions and Expositions

National Association of Practical Refrigerating Engineers: Annual convention, Sherman Hotel, Chicago, Ill., Nov. 1-4. Secretary, Edward H. Fox, 435 N. Waller Ave., Chicago.

American Society of Mechanical Engineers: Annual meeting, Dec. 5-9, New York City. Secretary, Calvin W. Rice, 33 W. 39th St., New York City.

National Power Show: At Grand Central Palace, Dec. 5-10, New York City. Managers, Charles F. Roth and Fred W. Payne, International Exposition Co., Grand Central Palace, New York City.

American Society of Refrigerating Engineers: Annual meeting, New York City, Dec. 7-10. Secretary, David L. Fiske, 37 W. 39th St., New York City.

National Warm Air Heating Association: Meeting, Dec. 7-8, Urbana-Lincoln Hotel, Urbana, Ill. Managing Director, A. W. Williams, 3440 A. I. U. Bldg., Columbus, Ohio.

American Society of Heating and Ventilating Engineers: Annual meeting, Jan. 23-25, Hotel Gibson, Cincinnati, Ohio. Secretary, A. V. Hutchinson, 51 Madison Ave., New York City.

New German Book on Drying

"DIE TROCKENTECHNIK," by M. Hirsch, Published by Julius Springer, Berlin, 1932.

For many years the classical work on drying has been "Drying With Air And Steam," by E. Hausbrand, published in Berlin and first translated into English in 1901. It is confined mainly to a discussion of the quantities of heat, air and steam needed for the removal of water under various conditions of temperature, pressure, air motion and humidity. While the tables and charts furnished were quite complete it found limited use in America due to the failure of the English editions to use the English system of weights and measures. It is to be hoped that M. Hirsch will not only publish an English edition of his new text but also that it will be planned to meet the needs of American engineers.

This work is not only very well illustrated but it is international in character as it draws on, and refers to, the technical literature of France, England, the United States and Germany. It is therefore, far more valuable than a text confined to the practice of a single country.

An interesting feature is the absence of the usual psychrometric charts based on atmospheric pressure. In their place are used the *i-x* diagrams for air-water vapor mixtures by Mollier. Such diagrams are particularly useful in solving problems in which the pressures are above and below atmospheric.

In a masterful manner this book first discusses the physical-chemical side of drying problems thoroughly and then proceeds to show how these problems can be solved through the use of the *i-x* diagram. Three large removable charts are provided for this purpose. The second portion of the book is devoted to a discussion of the preparations necessary to carry out the drying program, the generally accepted practices in this field, the typical equipment used and the methods of operation of such apparatus to obtain best results.—*Malcolm Tomlinson.*

• • • • MODERNIZE

Heating, Piping and Air-Conditioning Systems FOR EFFICIENCY AND ECONOMY • • •

The Air-Conditioning System

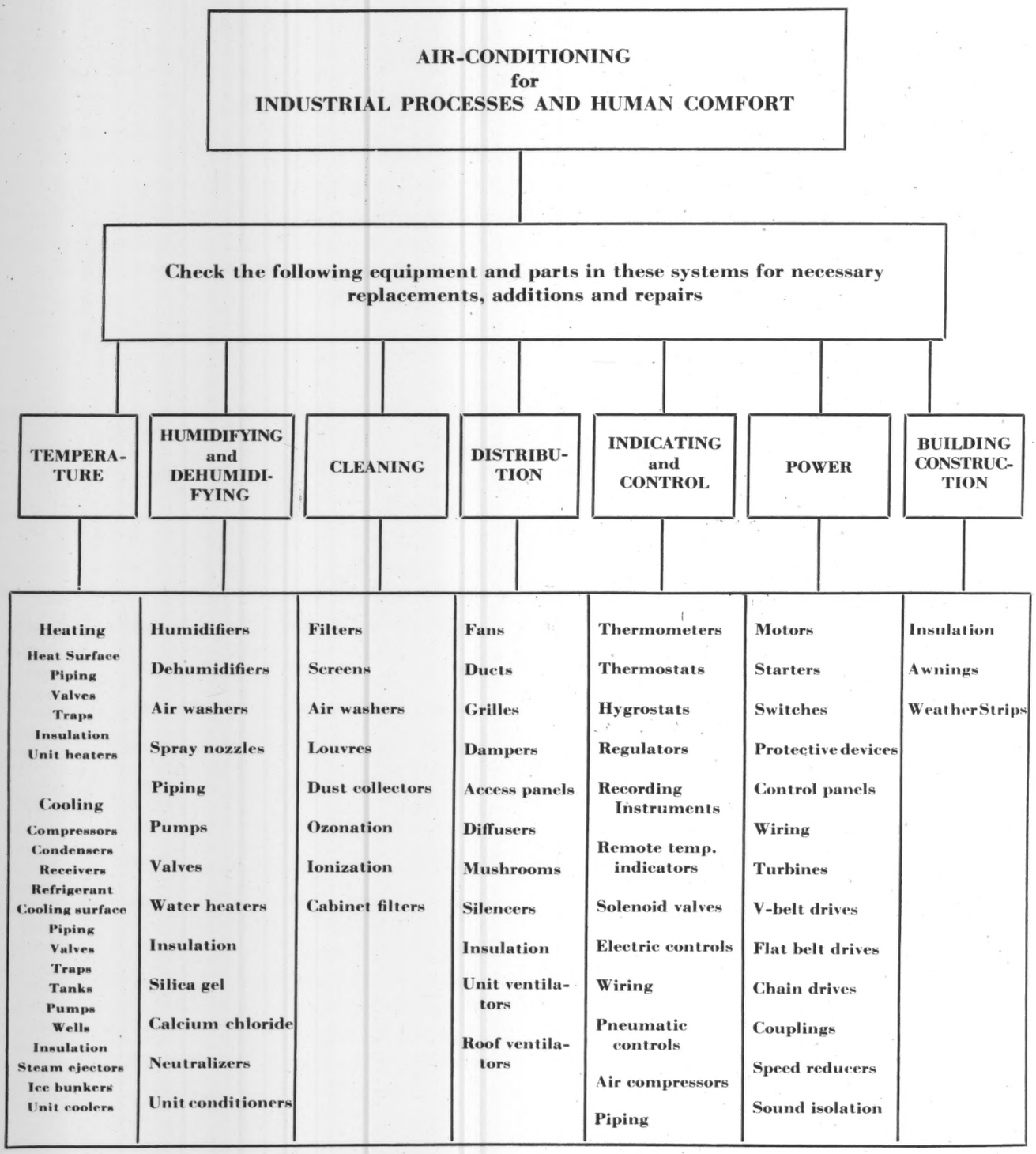
Check-List Aids
Proper Maintenance

Air conditioning for industrial processes and human comfort has developed to its present importance because in so many cases it can definitely pay good dividends on its cost through improved production or increased profits. Over three thousand industrial plants and hundreds of theaters, office buildings, restaurants and public buildings today are air conditioned.

From a modernization viewpoint, air conditioning represents two divisions of opportunity. In those plants and buildings where it is already installed it may be improved through the adoption of recently-developed equipment and through more careful maintenance of present equipment. In those industrial plants where it is not now installed it may be the answer to present production problems. In those commercial and public buildings where it is not now installed it may be the means of stimulating trade or increasing employe efficiency.

Air-conditionining systems are made up of the various pieces of equipment listed in the check-chart on the next page, each requiring the maintenance attention itemized on the page following. The check-chart and maintenance guide are given the engineer to facilitate his job of operating his air-conditioning system in the most efficient and economical manner. At the same time, they suggest that the engineer will do well to study the many possible applications of air conditioning where it is not now installed, both in plants and buildings.

Modernize Heating, Piping and Air-Conditioning Systems for Efficiency and Economy—Check-Chart for Air-Conditioning System



Check the equipment and parts which need attention. Replace with new, modern equipment where replacements will benefit; repair where repairs will suffice.

Maintenance Guide for Air-Conditioning Systems

Check the chart on the opposite page for equipment and parts which require attention. Find below suggested maintenance and repair operations:

Compressor

Indicate compressor to discover points needing attention?
Examine valves, spring tension; grind seats?
Clean and flush out water jackets?
Check piston and clearances?
Adjust or replace piston rings?
Examine piston rod and packing?
Check cross-head and guides for wear and operation?
Connecting rod, crankshaft, pins, bearings?
Lubrication? Charge with new oil?
Is compressor proper type and size?
Is replacement necessary?

Motor

Synchronous—inspect bearing adjustment by checking clearance between rotor and stator (shaft ten per cent above center of stator)?
Blow out motors with air and paint with spray coat of waterproof insulating varnish?
Commutators smooth and clean?
Voltage at motor proper? Feeders?
Replace brushes and springs?
Inspect bearings for oiling; remove sediment?
Take up bearings?
Install sound-isolation base?
Are motors proper type and size?
Is replacement of motors necessary?

Control

Inspect switchboards, panel boxes, motors and starters?
Replace arcing tips if burned?
Inspect pneumatic control apparatus?
Would automatic control simplify operation?
Would remote control facilitate operation?
Is complete replacement necessary?

Refrigerant Piping

Drain discharge oil trap, clean inside and out, and paint?
Pump out suction liquid separator, blow out any grease (heat by pumping back to thaw grease)?
Thaw out grease and oil in recirculating liquid line and blow out?
Check and replenish refrigerant?
Is replacement necessary?

Condenser

Atmospheric type
Blown down and cleaned inside?
Clean exterior surface of scale, dirt and corrosion?
Examine for leaks, tap with hammer for thin walls, install new pipes?
Level up troughs for even water distribution?

Double pipe, tubular or shell
Remove heads or bends, furnish new water gaskets?
Clean inside of tubes with scale cutter and wire brush?
Tubes and joints in tube sheets tested for leaks?
Clean shells inside and out; paint?
Adjust valves for proper water distribution?
Is replacement necessary?

Receiver

Drain oil?
Test for leaks?
Repack gage glass and cocks?
Clean and paint shell?
Is replacement necessary?

Direct Expansion Piping

Defrost evaporator surfaces?
Drain oil and refrigerant, blow out with steam and water or air to remove grease, oil, rust and scale from inside?
Examine pipe, valves and fittings with soapsuds and air pressure for leaks?
Apply coating of paint or rust-resistive material?
Pump out, exhausting air and foul gases?
Is replacement necessary?

Brine Piping

Examine for corrosion?
Test brine for acidity and alkalinity; adjust?
Drain piping and flush with water?
Clean strainer screen?
Clean piping, valves, and fittings with wire brush, test, and paint?
Is replacement necessary?

Water Piping

Drain and flush?
Clean all nozzles and strainers?
Clean and paint pipe, valves and fittings to retard corrosion?
Check water for corrosive properties; install equipment for treatment?
Cold water storage tank?
Is replacement necessary?

Brine and Water Pumps

Dismantle and examine impellers for corrosion and wear?
Inspect seal rings and bushings for clearances and wear?
Take up bearings; wash out?
Shafts in stuffing boxes; turn up if scored badly. Install new shaft?
Paint pumps and bases?
Pump packed too tightly?
Is replacement necessary?

Pipe Covering and Insulation

In good condition?
Paint with waterproof paint?
Fix loose insulation?
Is replacement necessary?

Gages

Proper size and style?
Check with dead weight?
Piping clean?
Install recording gages?

Air-Conditioning Maintenance Guide—Continued

Check thermometers for accuracy?
Check thermostatic controls, relays, cut-outs, alarms, etc.?
Clean thermostats. Test gaskets and diaphragms?
Is replacement necessary?

Fans

Remove suction pipe collar and examine wheel?
Scrape and brush blades of wheel and arms forming spider to remove dirt and products of corrosion; paint?
Inspect bearings and drive?
Install sound-isolating base?
Variable air control?
Is replacement necessary?

Air Washers

Check pressure on spray heads; clean?
Clean corroded parts?
Paint interior of washer (be sure surfaces are dry)?
Replace nozzle orifices if enlarged?
Clean and paint eliminators?
Insulate air washer?
Is replacement necessary?

Filters

Would filters save heat or recover valuable dusts?
Set up schedule for regular cleaning?
Check resistance to air flow; provide resistance gage?
Inspect motors and bearings; grease chain drives where exposed?
Remove sludge from reservoirs; replace oil?
Air intake in proper location so excessive dust and fumes are not drawn in?
Examine dirt screens?
Is replacement necessary?

Ducts

Clean out accumulated dirt?
Outlets and inlets arranged for proper distribution?
Dampers in good working order?
Fire dampers installed between floors?
Install sound-isolation material?
Insulate ducts?
Proper material?
Turning blades to reduce elbow losses?
Is replacement necessary?

General

Building insulated; awnings?
Zone control of system?
Roof of proper type to cut refrigeration load?
Roof condensation prevented?
Windows of right type; lock shut?
Low humidity rooms vapor-proofed?
Exhaust fans provided?
Stacks to remove local heat?

Unit Humidification

Water pressure O. K.—install tank?
Air pressure O. K.?
Controls clean and in good order; in proper locations?
Temperature maintained properly?
Air distribution uniform?
Heads cleaned regularly?

Choose Motor to Meet Conditions

By Francis A. Westbrook*

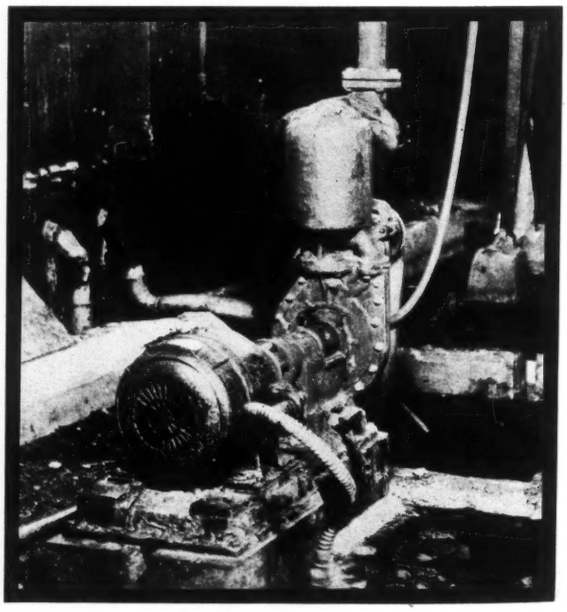
IN SELECTING motors for various services it is important that careful attention be given to choosing the type which will meet local conditions to the best advantage. This is well illustrated by the experience of the Sullivan Electrolytic plant at Kellogg, Idaho, where the same totally-enclosed, fan-cooled motor was first used to drive a fan handling acid fumes and after a year's service was installed to drive an electrolytic pump which delivers sump contents to circulation after spillage. To date no maintenance work has been done on the motor.

The severity of the conditions under which this motor operated when driving the fan is indicated by the illustration; the fumes ate the name-plate off. The solution handled by the pump which it now drives is 27 per cent sulphuric acid. The outside case of the motor has been given a coat of heavy asphaltum which, in the photograph, looks like acid holes in the metal but which protects it from corrosion.

This motor operates at full load 24 hours a day, day after day. It is believed that such continuous operation is the reason no trouble has been experienced and why none will develop.

The theory is this: In the first place, the windings and inner parts are protected against direct contact with fumes by being enclosed and the motor is kept at an even temperature by the fan cooling and the practically constant load. If the motor were to be stopped and started frequently, breathing would take place; that is, each time the motor became warm the air inside it would heat and part of it would escape through the bearing housing. When a motor and the air inside it cools a vacuum is created which would cause acid-laden air to be drawn into the motor through the bearing housing. This process, of course, would be slow but would probably eventually result in coil damage. With the motor running constantly it is believed that damage due to this cause will be postponed indefinitely.

*Consulting engineer, Center Conway, N. H.



Here are some notes made by one maintenance engineer, suggested by the steam-piping check-list published in the October issue

Renew 1½-in. brass piping to water column, No. 2 boiler. Pipe used at present is full of sediment.

Renew blow-off pipe line from No. 7 boiler. Simplify piping and use a more direct installation. Too many 90° elbows in present line. Fittings badly worn on inside.

Install 1-in. independent pipe line to remove the steam from boilers being taken out of service. This steam to be discharged to feed-water heater to save the heat. Make sure heater has suitable relief valve in good condition, in order that there is no chance of over-pressure accumulating in the heater when the high-pressure steam from the boiler is being discharged into the feed-water heater.

Repair steam leak in screwed joint below main whistle valve.

Use graphite and a suitable pipe joint compound on all joints replaced and repaired.

Repair flanged pipe joint on east end of main boiler steam header. Use suitable gasket cement; lubricate the bolts before applying the nuts.

Renew gaskets in leaking joints, as shown by separate list. Scrape gasket contact faces to remove all of the old gasket. Use suitable gasket cement. Lubricate all bolt threads before applying nuts. Tighten bolts evenly and carefully. Go over all flange bolts and tighten if necessary when steam is turned on line.

Replace roller pipe support under steam main in corner of tunnel. Present support in bad condition and out of position. Put new support in proper alignment and grout in place with cement.

Anchorage for 8-in. expansion bend in main steam line, Bldg. No. 4 is pulled out of place. Install new anchor bolts. Fill in with cement. Provide temporary bracing and leave in place until cement is thoroughly hardened.

Install new 8-in. steam expansion joint in heating system supply to Bldg. No. 9. Present expansion joint out of alignment and packing box gland broken.

Repack expansion joints on steam-hammer steam-supply line in forging shop.

Repair pipe insulation where conditions warrant.

Replace pipe insulation after repair work has been completed, steam turned on the pipe line and sufficient time elapsed to insure the repair work being properly tested.

Lubricate slip tubes on all expansion joints when steam is shut off the pipe line. Note. . . . When the steam is off the line the slip tubes naturally pull part of the way out of the packing box exposing the portion of the tube that slides in when steam is turned on. The lubrication is drawn in when the tube goes in position.

Install weatherproof jacket on pipe insulation of outside steam lines.

Cover large flanged pipe joints on main steam line, Bldg. No. 6.

Renew or repair leaking by-pass valves shown on separate list.

Repack valves leaking in valve stem packing box, as shown on separate list.

Reseat valves leaking, as shown on separate list.

Make list of spare valves on hand before arrangements are made to shut down the pipe lines for alterations and repairs.

Check over identification tags on valves used to admit steam for extinguishing fires in oil storage vaults, sawdust and shavings pipes, and lumber dry kilns.

Check all valves that should be operated periodically.

Tap valve body for ¾-in. drain on boiler feed pump discharge.

Replace leaking valves beyond repairs, as shown on separate list.

Steam traps should be closely inspected, necessary repairs and adjustments made to insure that the traps are in proper working order.

Move steam trap in Bldg. No. 5 from back of tunnel to a more accessible place near entrance.

Provide additional steam trap at steam-driven fire-pump throttle valve. At present steam supply line leading to this pump fills with the water of condensation and the temperature change causes the pipe joints to leak.

Inspect all thermostatic traps and make necessary replacements and repairs.

Install thermostatic trap between present float trap and main exhaust steam line, Bldg. No. 8. Thermostatic trap to be located and installed so as to discharge accumulated condensation to the atmosphere, in case of failure of the float trap to operate.

Repair pressure reducing valves as shown by separate list.

Check all pressure and vacuum gages for accuracy. Replace if necessary.

. . These notes suggest the method of use of the air-conditioning check-list and maintenance guide which appear in this issue.

Pipe
- Is material proper for the service?
- Is line the proper size to avoid excessive pressure drop?
- Are there too many turns in line?
- Is location and direction proper to permit maintenance and its best use?
- Is complete replacement advisable?
- Is additional piping needed?

Joints
- Screwed
  - Pipe compound?
- Flanged
  - Faces true and clean?
  - Gaskets?
  - Bolts tightened periodically?
- Welded

Support
- Are present means of support proper and adequate?
- Is complete replacement advisable?
- Is additional support needed?

Expansion Bends
- Proper type to care for the expansion?
- Adequately supported?
- Is pipe line anchored and guided correctly?
- Are additional expansion bends needed?

Expansion Joints
- Proper type to care for expansion?
- Is pipe line anchored and expansion guided correctly?
- Packing in good condition?
- Lubrication?
- Insulation?
- Is complete replacement advisable?
- Are additional expansion joints needed?

Insulation
- Was proper type and thickness originally applied?
- Is the insulation adequately protected from damage?
  - Exposed to dripping water?
- Is the insulation in good condition?
- Are flanges, tanks, and other equipment insulated where needed?
- Is complete or partial replacement necessary?
- Is additional insulation needed?

Painting
- Is piping or insulation painted for protection?
- Are piping and valves painted for identification?
- Is complete painting needed?

Valves
- Is proper type used in each case?
- Would remote-control valves simplify operation?
- Are valves accessible for maintenance?
- Are valves leaky?
  - Stuffing box need repacking?
  - Seats and discs in good condition?
- Are proper spare valves on hand?
- Are valves tagged for identification?
- Valves operated periodically?
- Valve body tapped for drain where needed?
- Is complete replacement necessary?
- Are more valves needed?

Steam Traps
- Air bound?
- Inlet line clear and discharge line open?
- Leaky valves?
- Trap mechanism bent or broken, pivots worn?
- Valve strikes seat squarely?
- Water-logged float or leaky bucket?
- Valve orifice, proper size and clean?
- Operating mechanism powerful enough; trap right size?
- Vent provided where there is considerable distance between a unit and a trap?
- Test cocks provided?
- Valves in by-pass leaky?
- Check valve needed?
- Dirt pocket needed?
  - Trap body filled with dirt?
- Is trap accessible?
- Is complete replacement advisable?
- Are additional steam traps needed?

Thermostatic Traps
- Thermic element broken or corroded?
- Volatile liquid missing?
- Handling water cool enough?
- Is complete replacement advisable?
- Are additional thermostatic traps needed?

Pressure-Reducing Valves
- Sized properly to avoid wire-drawing, chattering, etc.?
  - Use smaller valve in tandem?
- Does condensate accumulate ahead of valve?
  - Install steam trap to prevent?
- Control pipe arranged properly?
- Spring o. k.?
- Diaphragm o. k.?
- Seat o. k.?
- Stop valves to enable removal for repairs?
  - Install pass-by for hand control?
- Strainer or sediment trap?
  - Tee with clean-out on diaphragm chamber?
- Is complete replacement advisable?
- Are additional pressure-reducing valves needed?

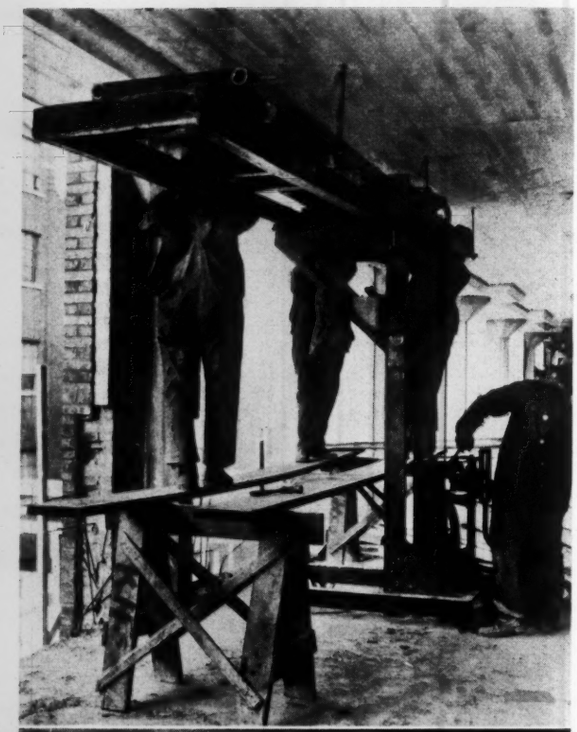
Indication and Control
- Gages checked for accuracy?
- Connections provided for portable meters?
- Gages and meters protected against damage?
- Connections to gages clean and unobstructed?
- Thermometers installed to indicate true temperature?
- Is complete replacement of gages and meters necessary?
- Are additional gages and meters needed?

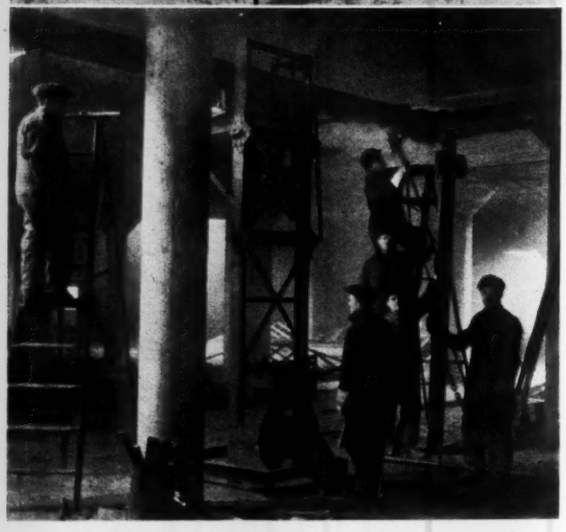
ON THE JOB

. . . a department devoted to short, practical articles on the installation, operation and maintenance of air conditioning, heating and piping equipment . . . . . .

Portable Hoists Aid Installation

. . . . *By* L. B. Murdock*





A PROBLEM in installing overhead equipment such as radiators, unit heaters, pipes, etc., lies in securing a safe and substantial means of supporting the load while it is secured in place; portable hoists may be used for this purpose. Their width must, necessarily, be small enough to permit free passage through door openings, while their height should be adjustable so as to clear them and yet enable lifting a load up to the ceiling. Loads weighing from 1,000 to 1,500 lb. must be handled, so the device used must be of substantial construction yet light enough in weight to permit ready movement from place to place.

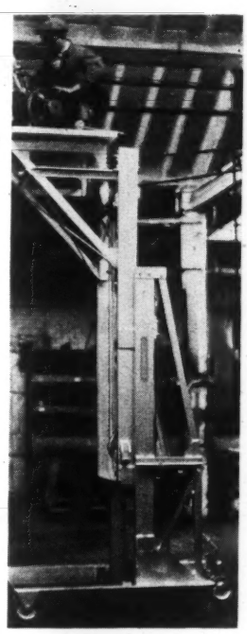
The solution to the problem as worked out by several heating and ventilating contractors who use portable lifters is illustrated in the views shown here. One illustration shows a 1,000-lb. capacity portable elevator raising overhead radiation into place. The lift is one of two used by a Chicago contractor. The elevating surface is 34 in. wide and 30 in. deep across which an auxiliary frame is placed when it is desired to raise an extremely long section of pipe or radiation. The base of the machine is 36 in. wide and 52 in. long; 7-in. diameter wheels in the base permit one man to move it about freely from place to place; it rests securely on a pair of feet when in operation. The top of the frame can be folded over to permit its passage through doors 6 ft. 6 in. high, although the machine can lift to a height of about nine feet.

Another view shows a similar machine used in installing a pipe line in a building of greater ceiling height. The lifting range of the hoist was increased three feet by inserting an intermediate section of that length between the main frame and the hinged top section. A similar piece of frame 18 in. long was also built for use with this machine when working under 11-ft. ceilings. While both the long and the short pieces can be used at once to provide a lift of 13½ ft., the load must be carefully distributed on

* Economy Engineering Co., Chicago, Illinois.

Above—1,000 lb. capacity portable elevator raising overhead radiation in place. Below—Similar machine equipped with intermediate section erecting piping

Telescoping portable elevator used in maintenance work on overhead equipment



either side of the frame when so employed, as the height thus obtained is greater than standard practice considers proper for a 36-in. width base, a machine of this height normally having about a 42-in. wide frame. In such service a narrow base often is an advantage; where the machines work in pairs, with the load supported between them, it is possible to use lifts as narrow as this one without detracting from their ability and safety.

Telescoping-type lifters are also available; the third view shows how one of these devices raises the load to the top of its main frame and then automatically extends itself to reach as much as 75 per cent higher in some instances. A sliding inner frame, fitted between the main structure and the lifting carriage, comes into use as soon as the elevating surface reaches the top of the permanent section. In this manner a 1,500-lb. capacity lifter with a height of 8 ft. 0 in. can raise material up to 12 or 13 ft. above the floor. The necessity of using intermediate sections is thus eliminated.

The manner of operation of both types is similar, the clockwise motion of the hand crank sending the platform up and the reverse motion lowering it. A brake holds the load in any position the moment the cranking pressure ceases.

It is not intended to portray this system of raising equipment as anything new or novel but merely to call attention to a method which seems to have undeniable merit, as evidenced by the fact that much of the piping in the new Chicago post office building was installed in this way.

Water Seal Protects Valve Diaphragm

. . . . *By* W. H. Wilson

A WATER seal is a help when fuel-oil pressure is used to control a diaphragm-operated valve. Oil often has an injurious effect on composition diaphragms; the water seal helps to prolong their life.

The loss of the diaphragm itself is not as bad as having oil blown about the pump house or otherwise scattered over the equipment. Also, when the diaphragm breaks the control is lost, which may cause trouble. A diaphragm breaking at night may cause large loss of oil. The sketch shows such a water seal. It is simply a water pocket or container located above the diaphragm chamber of the valve.

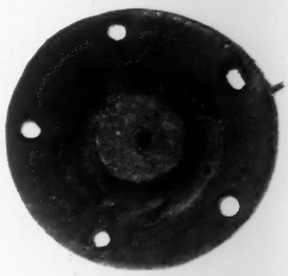
The container can be made by closing the ends of a piece of pipe, by welding or by threading and using pipe caps. All diaphragms, no matter how well protected, will deteriorate in time. Arrangements should be made so that the pipe connections can be conveniently removed and thoroughly cleaned before the new diaphragm is inserted.

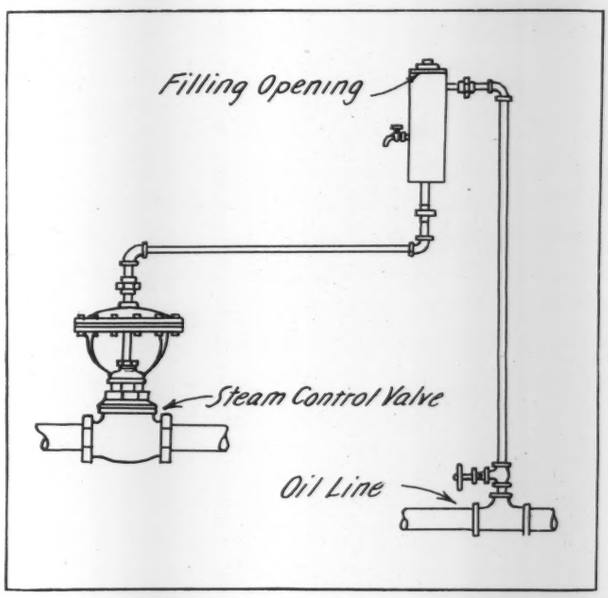
By disconnecting the unions, which should preferably be of the ground joint pattern, the cap of the diaphragm chamber, section of pipe, and the container can be easily removed and cleaned by steaming and washing with benzine.

The opening for filling with water is in the top of the container and is closed with a plug. A draw-off cock located in the side of the container provides a means of testing the water level and releasing the pressure, in case the shut-off valve leaks and it is necessary to disconnect and remove the piping.

The half-tone shows a ruptured diaphragm removed from the governor valve of a steam-operated fuel oil pump. The oil coming in contact with the diaphragm caused this trouble.

Right — Diaphragm removed from governor valve of steam-driven fuel-oil pump. Oil coming in contact with the diaphragm caused the trouble. Below—Water seal used to protect diaphragm when fuel-oil pressure is used to control diaphragm-operated valve





Steam from Seed Cooker Heats Water

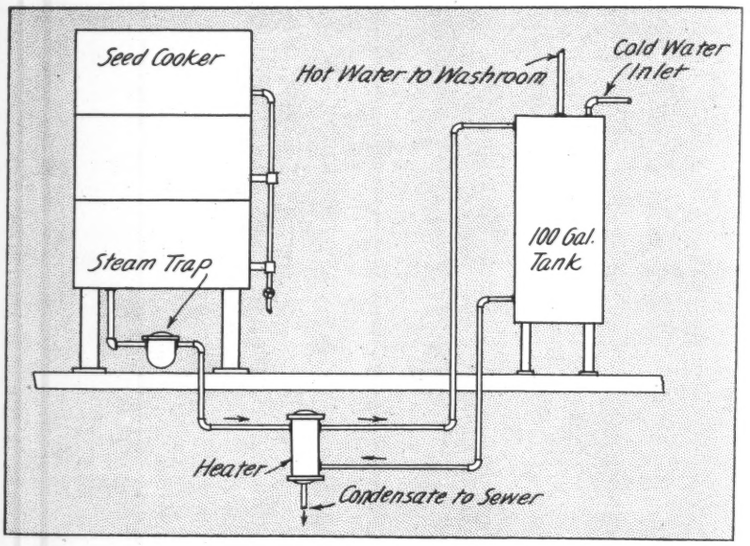
. . . . By J. Gordon Fletcher

LAST winter it was necessary to provide hot water for a washroom in one of our plant buildings where neither exhaust steam nor gas were available; it was not considered economical to use high-pressure steam for this purpose.

Steam was being used in four linseed cookers in operation continuously and it was for the men who operated these machines that the wash water was required. When the mill was shut down the hot water was not required. The steam traps from these cookers emptied into a sewer at a point directly under the washroom, the temperature of the trap discharge being 204 F as it entered the sewer.

A water heater was therefore installed in the trap discharge line below a 100-gal. storage tank and was piped to the water lines in the usual way.

Unlimited hot water has been made available by using heat formerly wasted to the sewer. Later the same idea was followed in connecting two steam-trap discharge lines through another water heater where a limited amount of hot water was needed.



Waste steam from four linseed cookers was used for heating water for washing, saving heat which formerly went to the sewer

The expense in each of these installations was very low and the results were most gratifying.

Saves 18% on Heating Cost

. . . . By H. G. Thomas*

SAVING eighteen per cent in heating cost is certainly worth-while in any building. Last year, a chief engineer I know accomplished this by a thorough check-up of the system in his charge and repair and replacement of the parts of the vacuum-heating system which the investigation showed needed attention. Any heating system should be properly maintained, but when my friend came on the job he found that his predecessor had neglected it until fuel costs increased every month.

After I had gone over the job with him, he wrote me the letter published below; it seems to me that the points he mentions are well worth the consideration of engineers responsible for the economical operation of a heating system in a large building.

"This plant sure looks different since I've got all the things attended to that you and I decided ought to be done. To make sure I haven't overlooked anything, I'll tell you what we've done and then if there is anything else you think of that should be checked up you can let me know.

"I had the men start at the boiler. They removed every bit of soot and dirt from the tubes, fire box, ash pit, smoke box and breeching. Two tubes were badly pitted so new ones were put in and some of the others rolled where they had shown signs of leaking. Two of the grate bars were burned and had been dropping coal into the ashpit so were replaced. The damper in the breeching had a broken shaft and would not work, so we put in a new shaft. The damper regulator had been out of commission for a couple of years so we cleaned it up, put in a new diaphragm and new chains and hooked it up properly to the dampers and fixed the dampers so they operated easily.

"I had the openings around the base cemented up where air was leaking into the ash pit and then fixed the covering where it had fallen off the boiler. We repainted the front and boiler covering and breeching so it sure looks like a new outfit. Then we cleaned and repacked the main valve, cleaned and reground the blowoff cock, cleaned and repacked the water supply cock and tested the check valves.

"That put the boiler end in A-1 shape so we started on the supply mains. First, that trapped pipe we found was corrected and we made sure that the rest of the main was properly pitched. Two of the flange unions showed signs of leak so we put new gaskets in them. Then we cleaned and tested the drip traps on the ends of the mains and fixed the covering that was in bad shape.

"Next we cleaned and repacked all the valves on the riser branches and checked over the run outs for trapped piping. The risers were all in good shape except for some loose covering, which was fixed.

"That brought us to the radiator valves, so we repacked all of them and put in new discs where needed and reground all scored seats. The radiators were all in good shape and

* Engineer, Warren Webster & Co., Chicago, Ill.

standing o.k. so we started on the radiator traps. We took all the thermostatic elements out and cleaned out the bodies of the traps thoroughly. Then we cleaned all of the elements and tested them before putting them back in the bodies. The return piping was all in good shape except for a few trapped arms which were corrected; we also re-routed that return from the northeast corner so as to get rid of that lift.

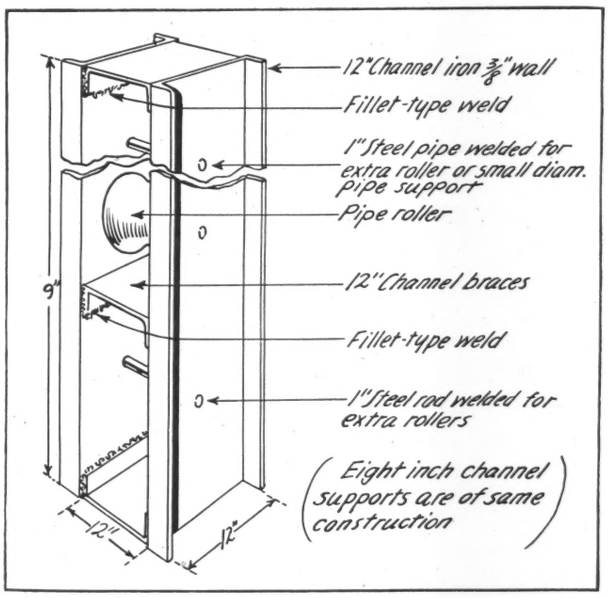
"Then we got busy at the vacuum pump, cleaned out the strainer and then cleaned and repacked the valves. Then we cleaned out the tank and pump casing, the runner and all the connections. We took the float control and cleaned it all up and made sure the float was o.k. Then we cleaned up all the automatic controls and oiled them, put in new gaskets and packing and cleaned and oiled the motor. Then we cleaned and packed the valve on the discharge line and tested the check valve.

"If you can think of anything else we might do please let me know. I'll write you later on how things work out."

Pipe Support Fabricated by Welding

THE accompanying sketch shows the design details of a typical pipe support fabricated by the oxy-acetylene welding process.

Extra heavy 12 in. channels with a ⅜ in. wall, and 8 in. standard channels were used, depending on the use of the support. The parts were cut to size with a blowpipe, and the various sections of channel iron



Pipe support fabricated by oxy-acetylene welding

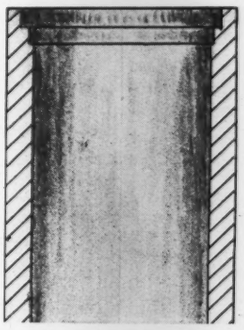
then assembled and welded together with fillet type welds. Steel rods 1 in. in diameter were then cut to the required length, inserted into position with pipe rollers on them, and welded to the vertical channels to act as axles for the rollers.

An 18-page pamphlet has recently been published by the committee on industrial gas research of the *American Gas Association*, 420 Lexington Ave., New York City on preparing industrial gas technique to carry a greater share of the industrial heating load. Research and investigational projects on various kinds of industrial heating processes are described.

Uses Pipe Caps in Repairing Air Washer

By Lawrence H. Georger

AN air washer had been installed several months and was apparently in good working order when spray back lash appeared in several deposits of water on the floor area before the air inlet. An investigation showed that in some manner eddy currents were the cause of the trouble; the difficulty was pinned upon metal supports which were used to support the stand pipes which carried the spray heads.

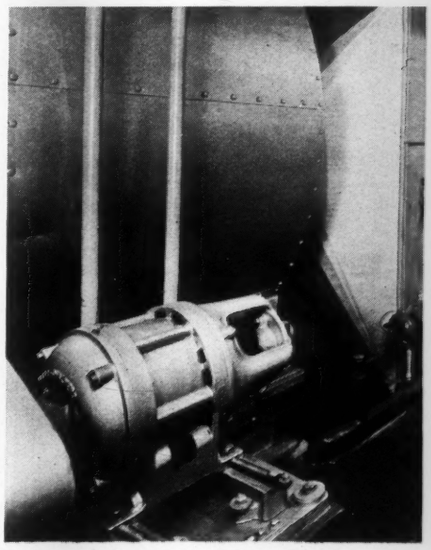


End of pipe counterbored ready to receive disc-type cap

Disc type pipe caps were inserted in the ends of the stand pipes. The ends of the pipes, being then held only by a shallow zee strip of metal, fitted closely to the roof sheet and eliminated structural parts which were the cause of the spray back lash.

The disc type of pipe cap used was a convex disc of steel or brass so designed that it would drop into the counterbored end of the pipe and expand to fill the opening.

The pipes were counter bored $\frac{1}{16}$ in. in a threading machine. The discs were then set in the ends of the pipe, pushed in place with thumb pressure, and set with a setting tool. A hammer blow on the end of the tool increases the diameter of the disc and causes it to engage the interior surfaces of the pipe; we have used this method for pressures up to 300 lb. The time required for capping the seventy-five pipes was 4½ hours.

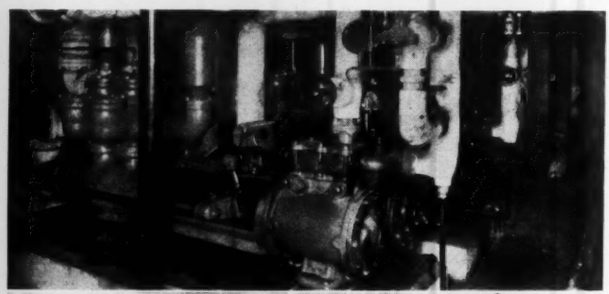


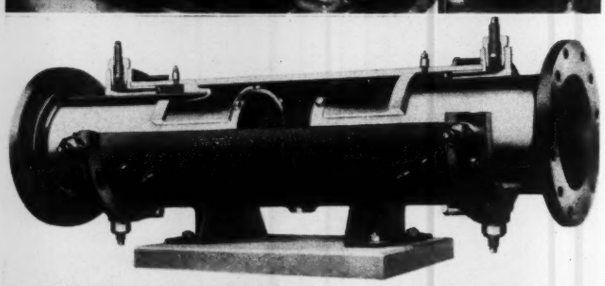
INTERESTING
Equipment.....
Developments

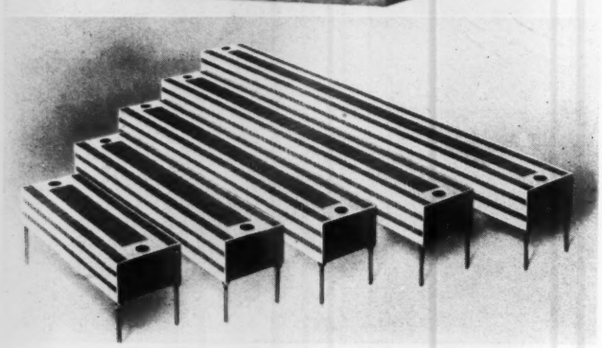
Pumps Handle Chemical Solutions

IMPROVED horizontal duplex steam pumps, especially designed and fitted for handling either water or chemical solutions, have been announced by the Worthington Pump and Machinery Corporation, Harrison, N. J. Built in three different types—valve plate, turret and pot valve—these units incorporate several features, among them being the use of stainless-steel drop-forged valves. Two of these pumps installed are shown at the top of the cut below.

The valve-plate pumps have the discharge valves screwed into a common removable deck. These units are of the submerged-piston type with all suction and discharge valves above the cylinders. Valve service is of the disc type, and all pumps have the standard duplex steam end.







For transfer, tank, circulating, boiler-feed and water-supply service, units of this group are standard fitted and have forced-in liners and screwed stuffing boxes. For collecting condensate returns from heating systems and returning them to the boiler, these pumps are furnished complete with receiver and automatic float control. For handling brine, caustic solutions and special liquids, the units are all-iron fitted; for handling acids and acid solutions they are bronze throughout. This group covers capacities from 5 to 320 g.p.m. at pressures up to 200 lb.

Turret-type pumps are available for capacities of 150 to 840 g.p.m. for discharge pressures to 75 lb. per sq. in. and with capacities of 100 to 500 g.p.m. for pressures up to 200 lb. per sq. in.

Pumps of the pot valve pattern, all with submerged pistons, come in three types: For pressures up to 500 lb. per sq. in. with a capacity of 40 to 96 g.p.m.; for boiler feed service, capacities of 15 to 760 g.p.m. and discharge pressures to 300 lb. per sq. in.; and for pressures up to 500 lb. per sq. in. and capacities of 140 to 1800 g.p.m.

New Method of Packing Expansion Joint

THE Yarnall-Waring Co., Chestnut Hill, Philadelphia, Pa., has developed a method of applying packing to an expansion joint which does not necessitate turning off steam in the line being packed, but can be accomplished through the application of plastic packing.

The principle of pressure lubrication is used with this company's cylinder guided expansion joint. This is accomplished by tightening screw-pressure guns in a cylinder built into the packing gland. These guns force a semi-solid fibrous packing into the stuffing box.

The cast steel gland is stationary, being bolted securely against the end of the shell of the expansion joint. Means for the application of plastic packing are provided by the use of two or more cylinder units (depending on the diameter of gland) the one shown having four cylinders for each gland. These fittings comprise a threaded stainless-steel plunger which operates inside a heavy cylinder seated in the rim of the gland. The plastic packing material is made of high grade long fibre

asbestos, combined with an inert material in which high fire-test mineral oil is stored. This packing is prepared in pellets, shaped to fit the cylinder.

When the plunger is screwed down, the packing is forced into the stuffing box through a channel in the body of the gland. Here an overhanging deflector vane distributes the packing. Because of these deflector vanes each piston and cylinder makes the packing move around the gland space in the same clockwise direction to insure uniform density of packing in the gland. When necessary to recharge, each cylinder fitting can be removed from the gland while under regular line pressure, without disturbing the action of the gland, as the deflector vanes act as check valves to prevent blowback. Recharging involves insertion of a new pellet of packing. Grease fittings are provided in the body of the joint in the packing space to provide lubrication for the moving sleeve, especially after long shut-downs. This system and packing are suitable for pressures ranging from subatmospheric to the highest pressures and temperatures used in modern steam-heating practice.

Fin-Type Convectors of Cast Iron

FINNED cast-iron radiators of the convector type have recently been placed on the market by the United States Radiator Corporation, Detroit, Mich.; rating tables for steam and water systems are available based on condensate output.

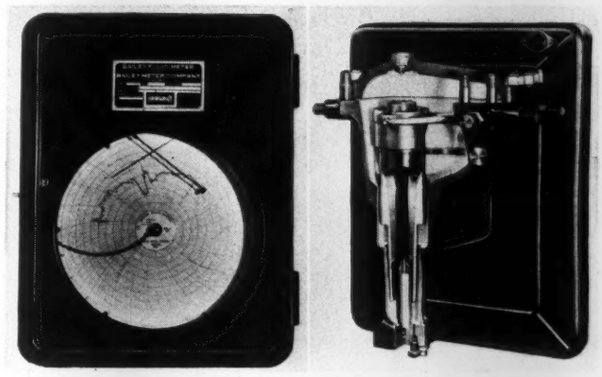
The radiators are made in depths of 3½, 5⅜, 7½, and 10⅝ in.; lengths are from 18 to 63 in. in 5-in. steps; all heights are 4½ in. The two smaller depth types have one centrally-located fluid chamber running the full length of the radiator and the other two sizes have two proportionally-spaced fluid chambers. The fluid chamber is designed to permit circulation and drainage without noise and its internal area is considerably in excess of that of the supply pipe. The fins are cast integrally with the fluid chamber and join together at their outer vertical edges forming the outer side walls of the radiator.

Several styles of incasements and enclosures are available for use with these radiators. Several of the radiators are shown in the view at the bottom of the opposite page.

New Fluid Meters Announced

NEW fluid meters for the measurement of steam, water, gases, and other fluids have been announced by the Bailey Meter Company, Cleveland, Ohio.

Indicating and integrating; recording and integrating; and indicating, recording, and integrating types are available. The illustration shows the recording and integrating meter; the record is made in lb. per hr., g.p.m., or any other desired units on a 12-in., 24-hr. chart. Accurate analysis of the record throughout the entire flow range is facilitated by the flow graduations being uniformly spaced; this also simplifies comparison with records of pressure and temperature which may be obtained on the same chart, when desired, by incorporating auxiliary recorders in the flow-meter casing. The total



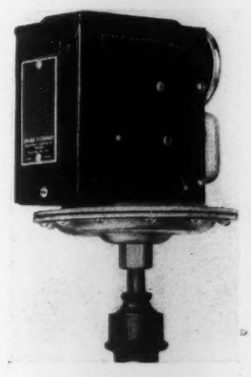
flow is given in pounds, gallons or other units by a 6-unit counter located behind a small window in the meter name-plate.

The construction of the meter is shown in the sectional view. The flow mechanism is operated by the difference in pressure produced between the inlet and outlet of the orifice or other primary element inserted in the pipe line. The inlet pressure is connected to the interior of the mercury-sealed bell, and the lower pressure is applied to the exterior of the bell. The bell responds to changes in the differential pressure, and moves up or down as a frictionless piston; it is shaped so that its vertical movement is directly proportional to the change in the rate of flow. Guide links direct the bell in a definite path, of particular advantage when the meter is used for marine service or as portable equipment. By means of a forked lever and spindle, motion of the bell is transmitted to the recording pen and integrator.

Permanent calibration is assured by protection against sudden surges or reversals of flow. The cases are arranged to permit mounting on the surface of a wall or panel, or flush on a panel. Auxiliary recorders, such as for temperature, pressure, vacuum or draft may be installed in the case. Meters are available for use under any commercial operating pressure.

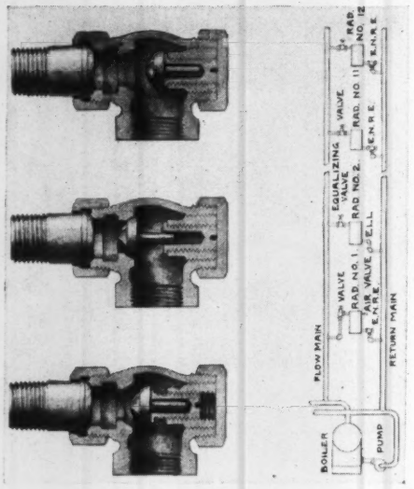
High-Water Alarm for Tanks

A NEW high-water alarm has been added to the line of pressure, float, and vacuum switches manufactured by the Square D Company (Industrial Controller Division), Milwaukee, Wis. It is for use over sump or open tanks and sounds a warning when the liquid reaches a dangerously high level.



The alarm consists of a single-pole pressure switch, a transformer, and a signal bell, wired in one unit and mounted on a 1-in. pipe. A rising column of liquid in the pipe exerts air pressure on the diaphragm of the switch, closing a contact and ringing the bell. The signal continues until the high liquid level is reduced to a safe limit.

Equalizes Hot-Water Systems



A NEW device for use in forced and gravity hot-water heating systems has recently been designed by Erwin L. Weber, consulting engineer, Seattle, Wash. It is an equalizing non-reflux elbow, used for balancing the flow of water to each unit of radiation and to prevent water from entering a radiator through the return elbow.

The device was originally designed for use in the Seattle Children's Home, wherein all parts of the heating system had to be made fool-proof and vandal-proof. It eliminates frequent draining and refilling of the system during the construction period.

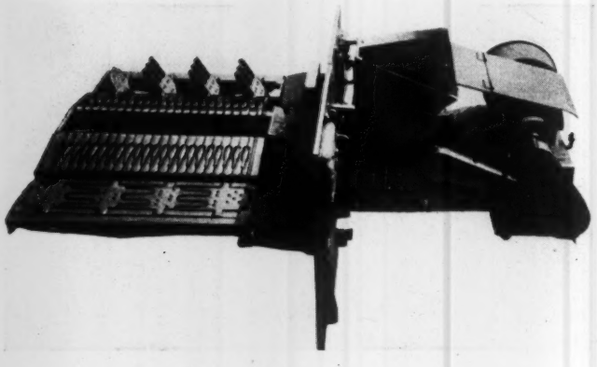
The device consists of a combination equalizing valve, shut-off valve and check valve, all in one mechanism, and is installed at the return end of the radiators, as shown on Rad's. 1, 11 and 12 of the diagram and marked E.N.R.E.

The three sectional views show (from top to bottom) the valve in its full open position, the check valve automatically closed after water attempts to enter through the return connection, and the mechanism in a partly closed position, for equalization of flow. This adjustment can be carried to a full closed position when it acts as a shut-off valve.

Existing poorly-balanced hot-water systems can be rectified by installing this device and performing the necessary balancing.

Underfeed Stoker of Ram-Type

A NEW underfeed stoker suitable for application to both heating and industrial boilers in the approximate range of 40 to 150 developed boiler horsepower has been placed on the market by Combustion Engineering Corporation, 200 Madison Ave., New York City; it embodies a number of distinctive features. Its fuel-feeding mechanism is of the ram or plunger type and comprises a main ram or pusher and auxiliary pushers, all mounted on a sliding plate which moves back and forth in the bottom of the retort. The grate surface is composed of stationary bars with moving elements or tuyeres which serve to agitate the fuel bed and make for effective air distribution. Standard equipment includes dead plates on either side of the active grate surface, but these may be replaced by shaking dump grates of sectional type.



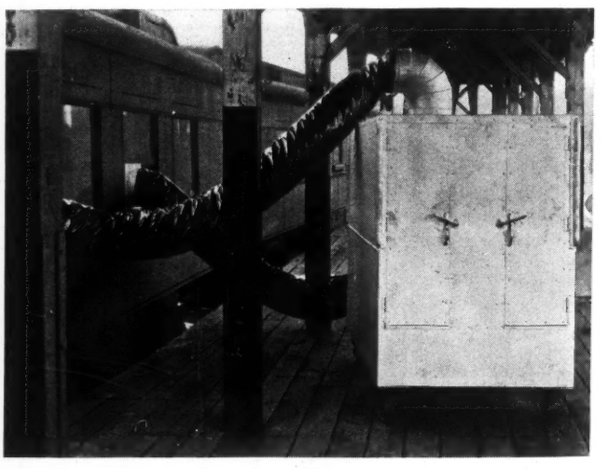
The drive is a constant-speed motor, variation in fuel supply being secured by a timing device which regulates the number of strokes of the plunger per minute in accordance with load requirements. An eight-to-one speed range is thus secured.

The fan is integral with the drive and is equipped with silent inlet dampers.

The control system, included as standard equipment, provides continuous adjustment of fuel feed and air supply over a wide range of load, and automatic on-and-off control during banking periods.

Railway-Car Precooler Announced

A NEW railway-car precooler which washes, cools, and dehumidifies has a capacity of 1,200 lb. of ice, with two ice chambers, each holding two 300-lb. blocks. The blocks stand on end and are supported by sloping grates, allowing the bottom portion of the forward blocks to be covered by the well water. Spray jets play on the ice, giving water at about 40 F. Between the two series of ice blocks there is an air chamber. At the end of this passage, the air divides into two side passages and is pulled over the ice and through the water sprays. At the forward end of the ice chambers are eliminator plates.



The blower then forces the air through waterproof canvas duct attached to a port at the blower end of the pre-cooler. The duct discharge end is secured to a metal window slide. If a car is pre-cooled before the entrance of passengers an inside duct may be attached to the in-

side of the window slide and laid on the floor in the aisle of the car.

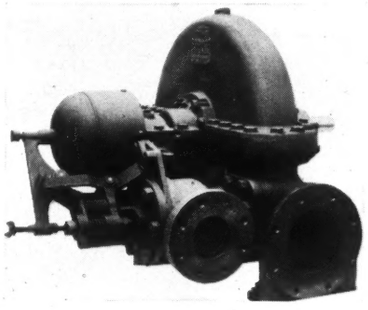
The air is recirculated by means of a canvas suction duct, the entrance port of which is placed on top of the pre-cooler near the blower end; the car end of this suction is attached to a metal window slide also.

The steel shell enclosing the equipment is aluminum painted.

The reservoir for chilled water is of sufficient capacity to allow for a two-hour run before draining. A drain is provided at one side of the port of the precooler operated by a valve. The precooler is iced from the rear through two insulated doors. The pump motor of 1/3 hp. and the blower motor of ¾ hp. are provided with safety switches. The motors are furnished for any desired current characteristics. The blower delivers 2,000 c.f.m. American Car and Foundry Company, 30 Church st., New York City, is the maker.

Turbine Drives Fans, Pumps

A NEW mechanical-drive, non-condensing steam turbine for driving centrifugal pumps, fans, and other rotating equipment has been developed by the General Electric Company, Schenectady, N. Y. This turbine is suitable for driving equipment at speeds from 1200 to 4000 r.p.m. and is available up to 250 hp. under suitable steam conditions.

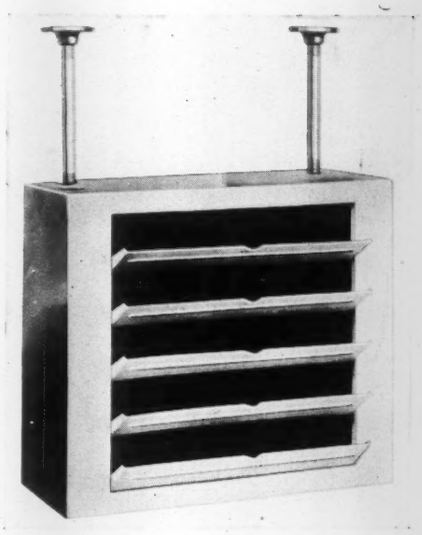


It is a single-stage machine with two rows of revolving buckets. The wheel casing is split horizontally to allow access to the internal parts and the steam and exhaust pipes are connected to the lower half of the machine. A centrifugal governor, with weights pivoting on knife edges to minimize friction, provides proper speed regulation. An independent emergency governor is included.

New Unit Cools or Heats

THE cooling and heating unit illustrated may be suspended from the ceiling for cooling, dehumidifying or heating such spaces as business or professional offices, restaurants, barber shops, beauty parlors, and similar establishments. The cooling medium is water which may be supplied from ice melting or mechanical refrigerating units.

A suitable hot water supply makes the unit usable for heating in winter. An electric fan and adjustable louvres regulate the distribution of air in the room after it leaves the coils. In new construction work, the unit may be built into the walls of the room which it is to heat or cool. The McCord Radiator and Mfg. Co. of 2587 E. Grand Blvd., Detroit, Mich., is the manufacturer.



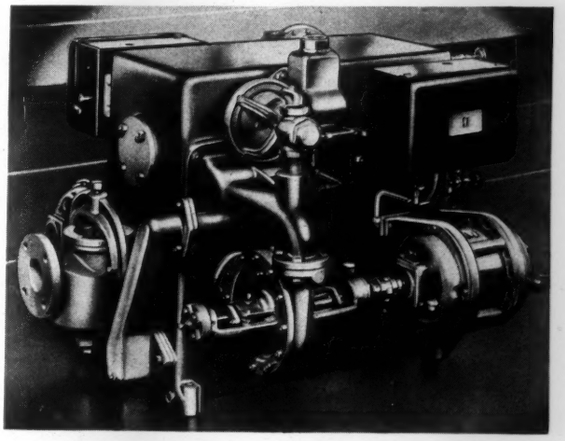
Tests Show Performance of Vacuum Pump

TESTS made on a new return-line vacuum pump reported by the Economy Pump Company indicated the following: When starting a cold job, the pump recorded an air capacity of 160 per cent of rating. The initial condensate reaching the pump after exhaustion of air was removed at the rate of 300 per cent of rated capacity. Thereafter, the removal of air and water was accomplished simultaneously.

At 160 F water temperature and 6 in. vacuum, pump capacity for both air and water was 100 per cent. At 195 F and 3 in. vacuum, pump capacity was 85 per cent for air and 100 per cent for water, the ability to handle water at this temperature being due to a jet-type vacuum producer, according to the report.

Both vacuum and condensation pumps of this company have been simplified and improved in appearance. They now employ rectangular receivers of cast-iron constructions; motors are of standard make, permitting quick replacement if necessary and all pump parts are readily accessible.

These changes, says the company, have been based on a study of operating conditions and are designed to improve operation, quietness and ability.



New Motor-Pump Unit

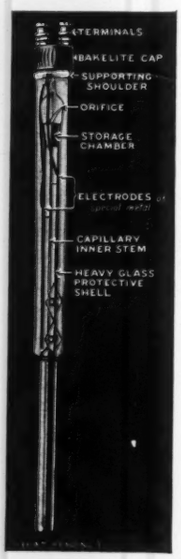
DESIGNED for use with air-conditioning apparatus, circulating hot water, cold water or brine, low-pressure boiler feeding, transfer of oils or other liquids, etc., the centrifugal pump unit illustrated here will pump against heads of 150 ft. or 65 lb. pressure in the single-stage types and twice that when furnished with two-stage unit in series. Capacities range from 5 to 200 g.p.m.

The unit consists of a single-stage, single suction, fully bronze-fitted pump with close-coupled motor. The impeller is of the enclosed type, mounted, keyed and locked on the extension of the motor shaft. The pump casing, of the volute type, is regularly of close-grained cast iron, but may be nickel-iron, bronze, monel metal, or other suitable alloy when specified. Stud bolts are so spaced in the casing to enable the discharge nozzle to be put in any one of four positions 90 degrees apart.

The manufacturer is the Lawrence Pump & Engine Co., P. O. Box 70, Lawrence, Mass.

Sensitive Thermo-Regulator

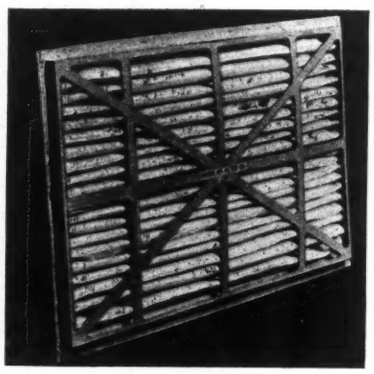
SENSITIVE to 1/10th of a degree throughout its range of 0 to 300 F, is the mercury thermo-regulator shown here, according to the manufacturer, Hiergesell Brothers, 2518 N. Broad St. Philadelphia, Pa. It is designed for controlling temperature of ovens, baths, refrigerants, rooms, etc., in conjunction with a relay.



The capillary and reservoir are filled with hydrogen to dampen the spark at the contact point; electrodes or contacts are of a special alloy metal. The relays are designed to operate within 12 milliamperes at 110 volts, 60 cycles through protecting condensers and resistors. The relay contacts will carry loads up to 30 amperes at 110 volts, a-c.

Dry-Type Filters Have Many Uses

THE dry-type air filter made by the Davies Air Filter Co., 39 W. 32nd St., New York City and illustrated here is applicable to air-conditioning equipment, warm-air systems, general ventilation, window and unit ventilators, air compressors, diesel engines, etc. The filter units fit into steel frames which hold any desired number of units; the units are removed by opening hinged doors in the frame and thrown away when dirty. Size



of a unit is 24 in. by 13½ in. by 1 in. and the effective area is 8 sq. ft.

The following data are given by the manufacturer:

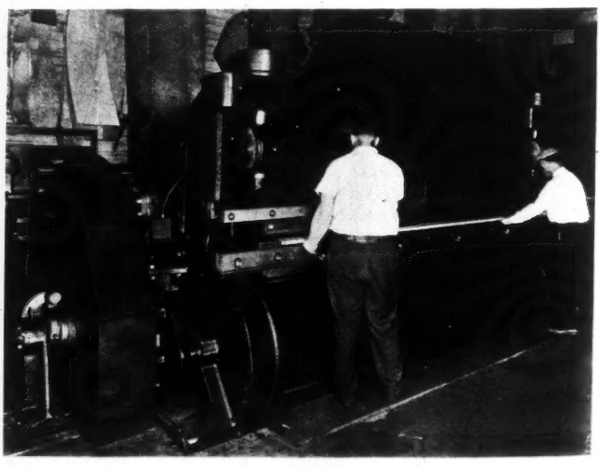
Volume, c.f.m.	Resistance, w.g.	Velocity through filter, f.p.m.
200	0.05	25
300	0.10	37
400	0.17	50

Press-Brake of Welded Construction

A NEW press-brake built by Farrel-Birmingham Company, Inc., Ansonia, Conn., is of welded construction. The bed is made of rolled steel plate, welded into a heavy girder section designed to bring deflection under capacity load to a minimum. Gear casings and housings for eccentric and gear shaft bearings are welded integrally with the bed.

The ram is solid rolled steel plate. Adjusting screw housings, guide bosses, etc., are formed by welding and the ram is annealed and normalized, as is the bed. To lessen over-all height, there is no driving mechanism on the upper part of the machine; it is all attached to or contained within the bed.

A feature of the machine is the principle of operation which pulls the ram from below. Power is transmitted from the driving motor to the flywheel on the drive shaft through a multiple V-belt and friction clutch. Gears enclosed in oil-tight casings connect the drive and intermediate shafts with individual eccentrics at each end. These, in turn, actuate the long, C-shaped connecting rods which pull the ram into the work.



Piping Air for Pipe Organs
Presents Interesting Problems

By . . . Lester C. Smith*

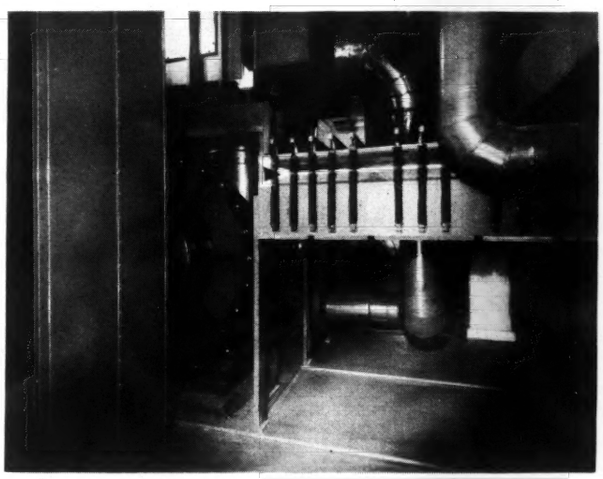


Fig. 1—Blower room in a conservatory showing 60-hp. main blower with multipanel filter in left foreground. Low-pressure regulator and muffler at right

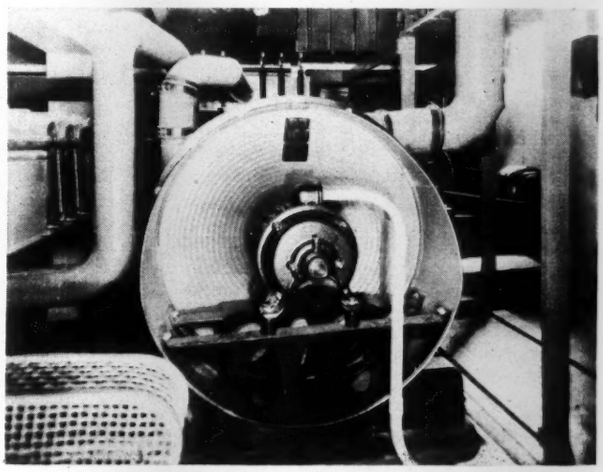


Fig. 2—10-hp. booster blower in foreground. Note V-belt drive for key action generator and mass of piping and blower-room regulators for each pressure

A PHASE of pipe-organ installation of particular interest to the engineer is the generation and piping of "wind" (using organ builders' parlance) up to the time it enters a pipe chest and is delivered to the speaking pipe. An installation which embodies nearly all of the features to be discussed in this article is that in a conservatory in Pennsylvania.

The blowers which supply the organ consist of a main machine of 60 hp. capacity, running at 870 r.p.m. Its rated capacity is 11,875 c.f.m. and it delivers air at pressures of 8, 12, and 15 in. of water from three different outlets. There is also a 10-hp. booster blower running at 1150 r.p.m. which draws its air from the 15-in. pressure outlet on the main blower and delivers air to the organ at three higher pressures—20, 25 and 30 in. of water. A third machine completes the installation; it is a 2-hp. exhauster which furnishes 12 in. suction for the console and duo-art equipment.

The main conductor, which carries the 8-in. wind, is 30 in. in diameter and is made of 16-gage heat-treated copper sheets. Other conductors range down to 6 in. diameter and are also of copper, though of lighter gage.

Air supplied from the blower room is drawn from the organ chamber proper, giving a recirculating effect. Before the air enters the blowers, it passes through an air filter of the multi-panel self-cleaning type which holds the operating resistance constant at all times. The filter also has a very conservative rating so that the resistance through it is held very low and the output of the blowers is unhampered.

*Assistant chief engineer, The Spencer Turbine Company, Hartford, Conn.

The blowers are located in a room of ample size one floor below the level of the floor of the organ chamber itself. Working on the theory that by overcoming the reverberation in the blower room, there would be less sound escape from this room, the walls and ceilings have been covered with insulation material with effective results. The blowers rest on an isolated concrete foundation separated from the other floors by heavy sections of cork.

Organ Blower Requirements

Organ blowers of the type mentioned are of the centrifugal type consisting of two or more stages, depending upon the air pressure to be delivered. They are designed to deliver an essentially constant pressure over their entire working range and this pressure should be at least 1 in. of water greater than the pressure on which the pipes in the organ are voiced. In organ blowing, the load is a constantly varying one; thus, the load on the blower motor may change from 25 or 30 per cent of full load to 125 or 130 per cent in an exceedingly short time. Again, the air load may diminish equally as rapidly and, of course, a pipe-organ blower must be designed to carry this rapidly varying load without any lag or other difficulties. It is essential that the proper volume and pressure be instantly available in all parts of the instrument whenever the organist demands it.

It is customary to install the blower in the basement, somewhat removed from the organ chambers themselves. It should be sufficiently removed from the organ proper so that any slight noise is not conveyed either through

the walls of the building or along the piping. Too much emphasis cannot be made on the importance of providing a suitable location for the blower. If possible a room should be provided for the blower where no other piece of equipment is installed. Doorways leading to this room should be amply large so that the blower can be set in position without being dismantled. As blowers are carefully balanced statically and dynamically to insure freedom from vibration and quiet operation, when machines are disassembled on the job, there is danger of upsetting this balance.

Special foundations are not necessary for the blower; neither should they be bolted or fastened to the floor. A good concrete floor or block is entirely satisfactory as a foundation and when installed the blower should be placed on the insulating pads (usually hair felt or cork) furnished by the manufacturer.

The room containing the blower should be clean and the air for the organ should be drawn from a clean place where the temperature is reasonably constant throughout the year. If a careful search is made, it is almost always possible to find several places in the basement of the modern church or auditorium where air for the blower can be drawn that is clean and of proper temperature. Some type of muffler on the air intake is advisable; usually a hinged door faced with felt or other sound-absorbent material serves nicely. This door should be hinged at the top and as nearly counter-balanced as possible so that the slightest negative pressure at the blower intake will cause the door to open with the resulting inflow of air.

It is customary to insulate the blower room with some sound-absorbing material so as to prevent transmission of mechanical noise. Fig. 3 shows a typical installation where the walls have been insulated. If these rooms are sealed up rather completely, some means must be provided so that the blower can draw in an ample supply of air. This is usually accomplished through a self-closing trap door in order that the door may open only a sufficient distance to permit the required volume to enter. Hence, since the intake air is usually drawn from outside the blower room, it is important that the air come from a clean, dry place of moderate temperature.

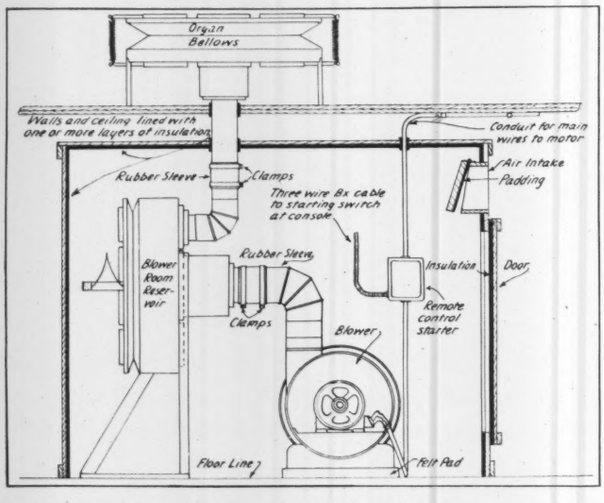


Fig. 3—Typical blower installation showing sound-proof enclosure

In small and moderate-size blower installations there is usually sufficient circulation of air through ordinary playing to ventilate the blower room effectively. Consequently, these machines may be placed in rooms which are completely sealed except for the single air intake. In large installations, however, additional means of ventilation should be provided. Motors and blowers 20 to 30 hp. in size generate a considerable amount of heat, even though running at normal temperature. If the room is ventilated only by the air delivered to the organ this heat is not dissipated but builds up in the room until ultimately abnormal temperatures are encountered. An effective method of ventilating the enclosures of large machines is to pipe the blower intake directly outside of the room and provide separate means for circulating air over the blower and motor. This circulation may be accomplished by radiation or by a separate ventilating fan.

If air filters are used, they should be connected to the intake of the blower by means of a flexible sleeve and a duct, rather than be installed as a part of the blower room wall. Usually filters are used only in large installations and where they are built into the wall of the room, there is not sufficient circulation to keep the blowing equipment at a moderate temperature.

In many installations, the so-called "recirculation" system is used. By this method two pipes are run from the blower room to the organ chambers. Air is drawn from the organ chamber to the blower room through one pipe and returned to the pipe chest through the other. With this method, air being drawn from the chambers is returned there so that both air and organ pipes are in equilibrium with respect to temperature. Since temperature affects the pitch of an organ pipe materially, it is essential that the air temperature be maintained essentially constant.

While this system has merits, it is the writer's opinion that equally good results may be obtained where the intake air is drawn from a clean place in the basement. In modern buildings, basement temperatures do not vary appreciably, and the expense of one wind conductor from the blower room to the organ chamber is eliminated. In addition the installation is apt to be quieter.

It has been suggested sometimes that intake air for the organ be drawn from the ventilating system, especially if the air has already been filtered. This plan should be avoided, however, since fan noise may be conveyed to the ventilating duct and thence all through the building unless extreme care is taken to muffle the pipes.

Conveying the Wind to the Organ Chest

The conductors or piping which convey the wind to the organ chest must be air-tight. This piping usually consists of galvanized sheet metal pipe with all joints and seams soldered tight. With the installation of these conductors fan or wind-rush noise must be given careful consideration. A column of air enclosed by a sheet metal conductor is a very efficient means for the transmission of sound; hence, this wind-way must be interrupted and broken up so that the pulsations and vibrations set up by the fans do not reach the chambers. Muffling of the wind column may be accomplished by

introducing an automatic gate in the pipe line directly at the outlet of the blower or by the installation at the same point of some baffle type of muffler.

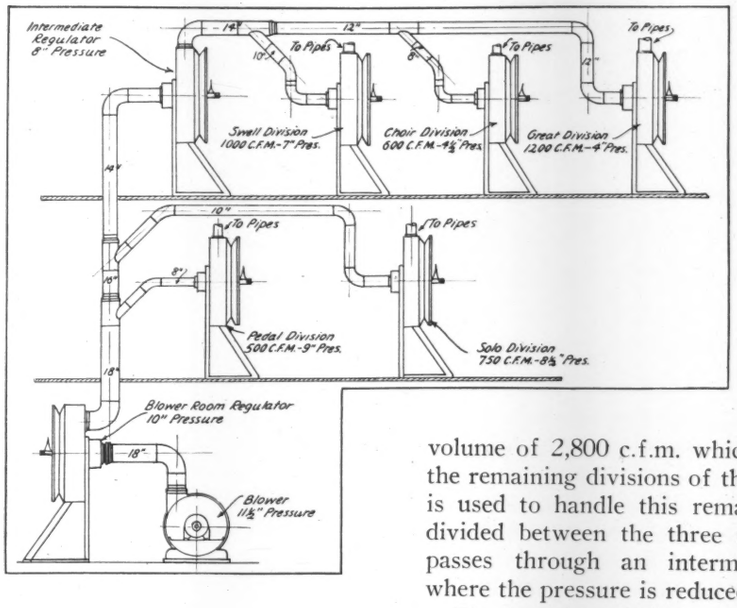


Fig. 4 — Schematic piping layout showing division of wind to the various voices in an organ

In the automatic-gate type the valve is customarily actuated by a bellows or reservoir so that it closes automatically as the air used by the organ decreases. With the straight baffle type muffler, the free area leading from the blower to the organ chamber is constant. Consequently where these are used, they should be designed with ample capacity and low frictional loss. The principle of the baffle is to convey the air through a tortuous passage and thus interrupt the sound waves. In making these turns and twists, however, there will be an appreciable frictional loss unless care is exercised to make sure that passageways are of ample size. The free area through them should be from three to four times the area of the pipe.

Air velocities up to 3,000 f.p.m. are permissible in the conductors leading from the blower to the organ chambers; in accepting this figure, however, it should be borne in mind that frictional resistance must also be given consideration. For instance, if a pipe line is 200 or 300 ft. long the pressure drop through such a run would be abnormal, especially for a blower supplying 4 or 5-in. (water column) pressure. Consequently, larger pipe and a lower velocity should be used. A total pressure drop not exceeding 1 in. is generally permissible in wind conductors, but a lesser drop is desirable when possible.

In many installations the pressure generated by the blower is the maximum required by the organ. For instance, the organ pipes may be voiced on 5, 7½ and 10 in. of water pressure. The blower supplies the maximum or 10 in. and the wind is conveyed through a single pipe to the organ. Here it is split up through pressure regulators and distributed to the various divisions of the instrument.

Consider the layout shown in Fig. 4. The blower is rated for 10-in. pressure and supplies an excess for safety sake of 1½ in., giving a total pressure at the blower of 11½ in. After passing through the blower-room muffler, it branches to the two high-pressure divisions in the organ. From there it passes through an intermediary step-down regulator and then on to the great, swell and choir divisions.

The blower is rated to deliver 4,050 c.f.m.; this is carried in an 18-in. pipe until the various organ divisions are tapped off. The pedal division draws 500 c.f.m. and the pipe extends from there 16 in. in diameter to the point where the solo division branches. This division draws 750 c.f.m. additional and there remains a volume of 2,800 c.f.m. which continues on and supplies the remaining divisions of the instrument. A 14-in. pipe is used to handle this remaining air and before being divided between the three remaining divisions, the air passes through an intermediate regulating reservoir where the pressure is reduced to 8 in.

The function of the regulating reservoirs or valves is to maintain the wind delivered to the speaking pipes at the desired pressure. In addition, they act as mufflers which serve to interrupt and smooth out the pulsations in the air stream. Frequently, these regulators are used as blower-room mufflers.

In extreme cases, where windage noise proves objectionable in other parts of a building than the auditorium, the conductor piping may be wrapped with some sound-insulating material, such as 1 or 2 in. of hair felt. In other cases, the wind piping is sometimes boxed and the housing filled with sand, ground cork or sawdust. Ordinarily it is not essential to wrap wind trunking.

Electrical Control of Blowers

Control for blowers is fully automatic. There is located at the organ console or key desk a push button which controls both the operation of the blower and the key action generator. Incidentally, the key action generator is customarily driven from the blower motor, either by means of a belt or by a coupling directly connecting the generator to the motor shaft on the opposite end from which the fans are mounted. If it is driven from a separate motor, the starter controlling this second motor should be tied in with the same push button at the organ console which controls the blower motor.

Motor control equipment should be simple. It is possible to start organ blower motors up to 30 hp., with the across-the-line type of magnetic contactor. This starter should be operated from the organ console by means of a momentary contact push button wired for three-wire control so as to provide true low-voltage protection. For a-c. motors, where the across-the-line type starter cannot be used, a reduced voltage starter of the resistance or compensator type may be furnished.

For d-c. equipment the time-limit acceleration-type starter with several steps of resistance may be specified. It should have at least three steps of resistance and be so timed as to bring the motor up to speed very gradually. Fifteen seconds is a good average time for the ordinary d-c. organ blower.

Heating · Piping
and Air Conditioning

EDITORIAL

In these days when waste and inefficiency are inexcusable it is often the duty of the engineer in charge of the heating, piping or air conditioning for a plant or building to bore through the lethargy or lack of engineering understanding on the part of his non-technical superiors and eliminate waste in spite of opposition. In other words, the engineer must sell his ideas on modernization and improvement to those who hold the purse strings.

It is stated that fifty per cent of the equipment in industrial plants is obsolete and that replacement of such equipment by modern equipment will pay a dividend sufficient to warrant its purchase. It has also been stated that the chief bar to such warranted purchases is lethargy, inertia and lack of engineering understanding of what can be accomplished on the part of business executives.

It is part of the object of many of HEATING, PIPING AND AIR CONDITIONING'S articles to make clear to engineers and executives not only how economies can be effected in heating, piping and air conditioning but also how this information may be presented to business executives in a clear, convincing and understandable manner. The man who knows what a rehabilitation program can accomplish for his own organization should be prepared to sell his ideas to his superiors, using his engineering analysis and such reports of the experiences of others. In this way, his experience and knowledge become of greatest value to his company.

Heating·Piping
Air Conditioning

1895

American Society of Heating and Ventilating Engineers

JOURNAL SECTION

NOVEMBER 1932

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Air-Conditioning System for Low Humidities Required During the Manufacture of Telephone Cables

By F. H. Kruger*, Chicago, Ill.
NON-MEMBER

TELEPHONE conversations probably do not suggest air conditioning. The maintenance of low humidities, however, is invaluable in the manufacture of telephone cables each of which is made up of a multiplicity of small wires insulated with paper and surrounded by a lead sheath.

There are two general types of telephone cable—loop and toll. Loop cable is used between the central office and the subscribers, while toll cable is used between distant points. Due to the length of the toll lines and the frequencies used, the transmission requirements for toll cable are more exacting than for loop cable. Since the characteristics of a cable current depend largely upon the conductance of the insulating paper, it follows that the moisture content which largely determines the magnitude of this property, must be kept at an especially low value for toll cable.

To obtain the desired low conductance in the cables, it is necessary to remove the moisture which has been absorbed by the paper in the preliminary manufacturing processes which are carried on in unconditioned rooms. This is done by drying the cores of stranded paper insulated conductors in heated vacuum dryers for a sufficient length of time prior to the application of the lead sheath. For best operation it is found convenient to keep a stock of cable cores available for the sheathing presses. It is this cable core storage space which must be maintained in a dry condition in order that there will be an exceedingly small *regain* of moisture by the cables (See Fig. 1). To maintain the low values of conductance in toll cable it is necessary that this room have a relative humidity not greater than 0.5 per cent at the usual operating temperature of 115 F. Since the transmission requirements for loop cables are not so exacting as for toll cables, relative humidities as high as 7 per cent are satisfactory in the room used for the storage of loop cable cores. Consequently it has been found economical to provide two storage rooms rather than maintain the exacting condition for toll cable cores throughout the entire storage space.

This paper considers the requirements of an air-conditioning system to maintain the necessary humidities and temperatures in the cable storage rooms. The selection, design and performance of a combined refrigeration and moisture adsorption system are described. A two-stage refrigeration system cools and consequently drys the air which is delivered to the adsorption system and to the loop cable storage room for the removal of heat. The adsorption system supplies air of a low moisture content to the toll cable storage room. Air recirculated from the toll room, maintains the correct humidities in the loop cable storage room. Silica gel placed in two beds or adsorbers, dehydrates the air passing through the adsorption system. An air heater and cooler are successively used to condition the moistened gel in the adsorbers alternately. Finally the distribution of air and the humidity determinations in the storage rooms are discussed.

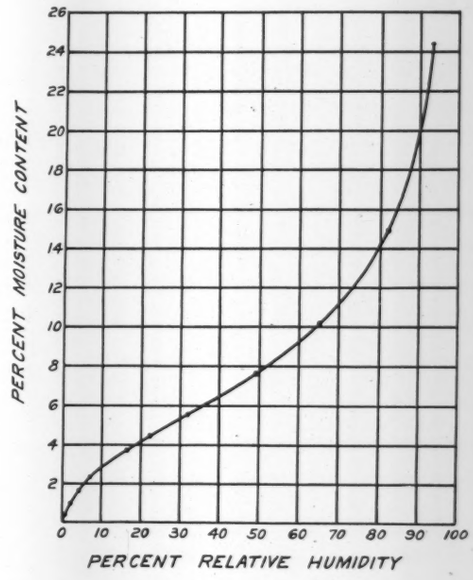

Fig. 1—Moisture content of cable paper in equilibrium with different relative humidities at 75 F

Design Requirements of Air Conditioning System

The toll storage room considered in this paper is 187 ft long and the loop storage room is 172 ft long. Both rooms are 30 ft wide and 11 ft high. The volumes of the toll and loop storage rooms are accordingly 61,710 and 56,760 cu ft, respectively. Fig. 2 is a view of the

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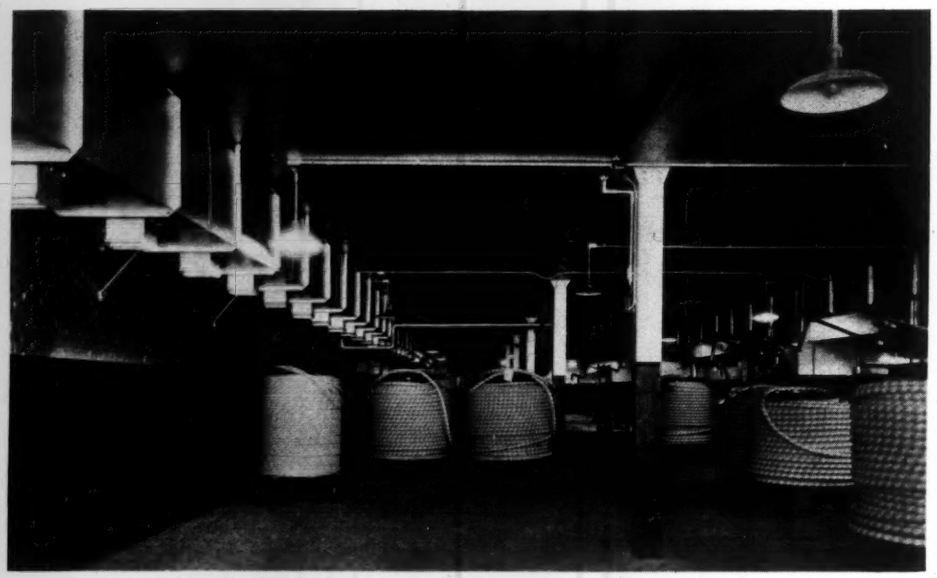


Fig. 2—Toll cable storage room with supply duct at left and recirculating duct and press room door vestibules at right

toll storage room. A large vestibule between the two storage rooms is used for the introduction of dried cables and the removal of core trucks from the toll storage room. Two vestibules between the loop storage rooms and the adjoining manufacturing room serve for the removal of core trucks. The walls are brick, 13 in. thick, the ceiling consists of 2½ in. of concrete and 1 in. of plaster and the floor is of concrete.

The first requirements of the problem of providing a suitable air conditioning system, of course, necessitated the determination of the volumes, humidities, and temperatures of the air to be furnished. The necessary volumes and humidities are dependent upon the water vapor to be removed from the rooms and the temperature required is dependent upon the cooling to be effected.

To establish the volume and humidity requirements of the air to be delivered by the conditioning system, it was necessary to ascertain the rate at which water vapor entered the rooms. By the extrapolation of data obtained on other low humidity rooms it was possible to approximate the entrance of water vapor into the two rooms. However, tests and calculations were made to check these values. In this connection it was considered that moisture would enter the rooms by:

1. Diffusion through the walls, ceiling and floor.
2. Diffusion through open communication doors and through cracks around the doors, and
3. Exchange of air when doors are open due to difference in density.

The infiltration through the walls, ceiling and floor of the toll room was determined by absorbing the water vapor in sulphuric acid placed in an air-tight test box sealed to the wall. A value of diffusion through the open doors was obtained by exposing sulphuric acid in air currents such as prevailed through the doors. The rate at which air entered the room due to the difference in density was determined by anemometer readings at doors of a similar room under similar conditions of static pressures and temperatures.

The tests described in the preceding paragraph showed the rates at which aqueous vapor entered the rooms and permitted a calculation of the amount of dry air that would have had to be supplied to dilute the incoming moisture to the desired value. Before providing an air-conditioning system of this capacity various methods were, however, investigated for reducing the entering moisture and it appeared from these studies that it would be more economical to make certain changes to reduce its rate of infiltration than to provide a conditioning system of the capacity shown necessary by the tests and calculations. A waterproofing lacquer suitable for the prevailing temperatures was accordingly applied to the walls and ceiling. A surface hardener was also used on the concrete floor. In addition all openings in the walls and around the doors were sealed and vestibules with an exhaust system were installed on the communication doors of the toll storage room. The infiltration was redetermined and the capacity of the air-conditioning system was based upon the net entering moisture and the required end conditions.

In order to insure the necessary cooling by the volumes of dehumidified air determined, certain limiting temperatures were established from tests and heat balance calculations. The heat losses from the rooms are small, whereas appreciable heat is supplied to the rooms by the hot, dried cable cores at an approximate temperature of

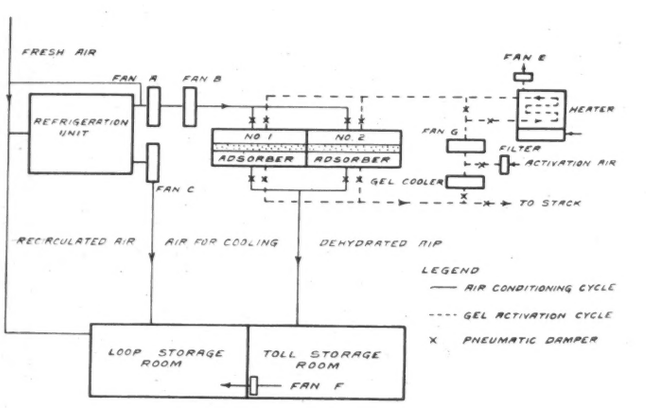


Fig. 3—Air flow diagram of complete conditioning system

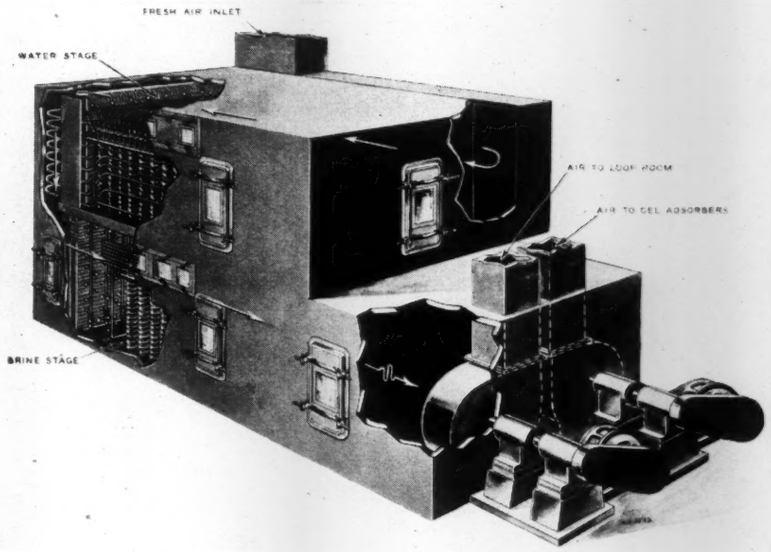


Fig. 5 — Refrigeration dehumidification unit showing evaporator coils, eliminators, spray nozzles, access doors and fans

230 F. These studies indicated that the cooling load required by these rooms was the removal of 4000 Btu per minute from the loop room and 3400 Btu from the toll room. The temperatures of the specified volumes were determined from these cooling loads.

Various arrangements were carefully studied for providing air at the proper inlet conditions to the room and a combination refrigeration and moisture adsorption system was found to be best suited when operating costs, flexibility and reliability were considered. The diagram of the air flow system developed as a result of these studies is shown in Fig. 3. In the refrigeration dehumidification unit, 6000 cfm of air can be partially dried for delivery to the silica-gel dehydration system and 3000 cfm can be sent to the loop storage room to maintain the desired temperatures. When cooling is required in the loop storage room air is supplied directly from the refrigeration unit.

Refrigeration Dehumidification System

Moisture is condensed from the air in the refrigeration unit by the removal of latent and sensible heat. This unit consists of two cooling stages—the water stage and the brine stage as shown in Fig. 4. Air is supplied to the plenum chamber of the water stage from the fresh air intake and a recirculating duct connecting the loop cable storage room. In the water stage the air is cooled to a dew-point temperature of 42 F by passing through sprayed water and between evaporator coils over which water is running. It then passes through the water-stage eliminators into the brine stage where it is cooled by a calcium-chloride-brine spray and similar evaporator coils to 32 F. After passing through the brine stage eliminators and unit-type filters, which remove any entrained droplets of brine, the air is supplied to the adsorption system by fans *A* and *B* and to the loop cable storage room by fan *C* as shown in Fig. 3. During the winter fresh air is supplied directly to the fans through a by-pass to the fresh air supply duct. This permits the delivery of air to the adsorption system and the loop storage room at a somewhat lower temperature than would be obtained if it were drawn through the entire dehumidifier.

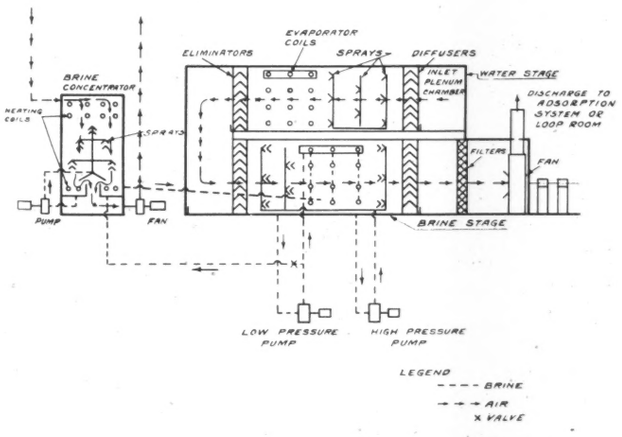


Fig. 4—Schematic diagram of refrigeration dehumidification unit showing air cycle and brine cycle; the water cycle is similar except concentrator is omitted

The two stages of the refrigeration dehumidification unit were built in a double-deck design as shown in Figs. 4 and 5 because the space available for this installation was limited. The housing of the unit is made of corrosion-resistant steel. Access doors equipped with observation windows are located at points where inspection is necessary. Two elongated doors of this type are at the top of each stage in order to permit adjustment and maintenance of the flooding pipes described later. The expansion, stop and safety valves were grouped together in an insulated chamber fitted with cold-storage type doors. A special type of seal prevents the water and brine in the dehumidifier from entering the valve chamber.

The water and brine used to cool the air are sprayed from nozzles assembled in three banks in each stage. The low temperatures of these cooling agents are maintained by one set of evaporator coils placed in the water stage and another set placed in the brine stage. Ad-

ditional capacity has been provided in the later stage to cool the air to 0 deg F in order to permit this unit to take some of the normal dehydration load from the subsequent adsorption system during repair periods. Carbon dioxide is used as a refrigerant in the system. The heat transfer from the cooling agents to the refrigerant is effected by flooding the evaporator coils from slotted pipes. One flooding pipe is placed above each vertical stand of evaporator pipes. The flooding pipes are of a nominal inside diameter of 1½ in. and contain milled slots on the upper side. The width of the slots is varied so that a uniform distribution is obtained through the length of the pipe. The flooder pipes are connected through cocks to a supply header on the aisle side of the dehumidifier. By adjusting the cocks, a

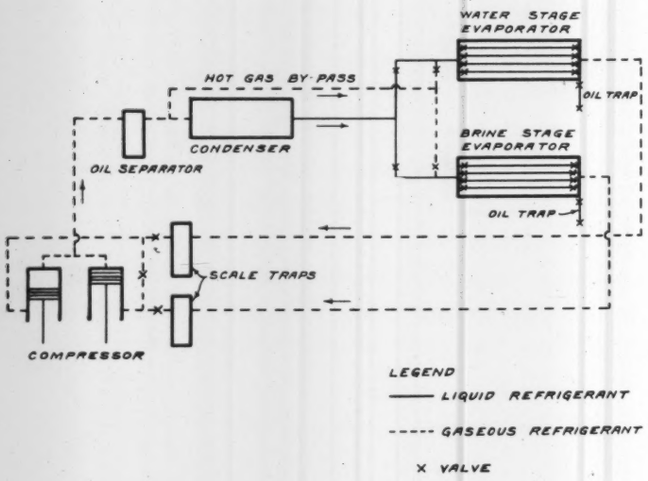


Fig. 6—Schematic diagram of refrigerant cycle

uniform amount of brine or water is supplied to each flooder. No clogging has resulted from the use of these flooding pipes. In case any pipes must be removed for service or adjustments must be made, the elongated doors previously mentioned are used for this purpose. A low head pump supplies the flooder pipes while a high head pump supplies the nozzles in each stage.

Two fans *A* and *C* are installed side by side in the outlet end of the dehumidifier. As previously mentioned, one of these fans delivers air to the loop storage room while the other delivers air to the adsorption system. To prevent air from being drawn into the housings along the fan shafts, they are equipped with glands to which pressure is supplied.

Under normal operating conditions with a dew-point temperature of 32 F, nearly 10 gal of water are condensed in the brine stage each hour. To eliminate the necessity for adding considerable quantities of calcium chloride to maintain the proper specific gravity of the brine, a small amount of brine is taken from the discharge of the low pressure pump and furnished to a concentrator as shown in Fig. 4. The concentrator is a spray-type dryer used to evaporate the water. It is of the down-draft design. During operation air from the fresh air inlet is drawn over heating coils placed at the top of the unit and is passed through a heated spray of the diluted brine from which the excess water is evaporated. This moisture laden air then passes through a set of eliminators to remove the entrained drops of brine and is drawn down into the fan chamber through a tubular opening in the annular shaped brine pan. The moist air is discharged by a fan through a stack to the roof. The brine is heated by steam coils placed in the pan and maintained at a constant temperature by an automatic temperature controller. A circulating pump supplies the spray nozzles and also agitates the brine in the pan.

The refrigerant cycle is of the conventional type as shown in Fig. 6. The liquid carbon dioxide from the condenser is passed through expansion valves into the water stage and brine stage evaporator coils where it is evaporated at desired reduced pressures. Each set of coils is connected to the compressors by a separate return or suction line equipped with a scale trap. Two cylinder double acting vertical compressors are used. The connections are arranged so that one cylinder of one machine may be operated on the evaporator coils in the water stage and the other cylinder on the evaporator coils in the brine stage, or so that both cylinders of any compressor may be operated on either set of evaporator coils. After the gas is discharged from the compressors at the necessary pressure, it passes through an oil separator and is liquefied in a condenser where the heats of vaporization and compression are removed by the cooling water. The compressors are directly connected to synchronous motors. A lubricating system, developed especially for these machines, supplies oil directly to the cylinders of the compressors, and to packings on the piston rods. To remove any viscous or congealed oil which may accumulate in the evaporator coils, a hot gas by-pass is provided between the discharge line and the liquid headers on the evaporator coils. (See Fig. 6.) By opening the expansion valves wide on a few coils at a time, hot gas is introduced to the evaporator coils and loosens the oil which has collected on the inside of them. The oil which has collected in the coils is swept out with the gas which is moving at a high velocity and caught in traps connected to the suction headers.

To obtain a continuous daily record as well as to make comparisons and checks of performance, a number of recording instruments are employed. Recording pressure gages are used on the water and brine stage suction lines and the discharge line. Recording thermometers are used to show the temperatures of the water going into and coming from the condensers as well as the temperature of the refrigerant coming from the condenser. Recording and indicating flow meters operating on the thin plate principle are used in the ducts supplying the loop room and the dehydration system. Manometers are permanently connected across the inlets and the outlets of the different fans.

Adsorption Dehydration System

As already stated, fan *A* and fan *B* of Fig. 3 which are in series, deliver air to the silica-gel adsorption system[1,2]. These two fans draw at least 6,000 cfm of air

[1] J. Wells and L. G. Wade (*Chemical and Metallurgical Engineering* 39:146-9 March, 1932.)
[2] J. Wells and L. G. Wade (*Power* 75:429 March 22, 1932.)

through the refrigeration unit, force it through the silica-gel adsorbers and distribute it in the toll storage room. In this dehydration process air at a temperature of 42 F above the gel bed and moisture content not greater than 29.9 grains per pound of air is dried to a moisture content of less than 0.5 grain per pound of air. This corresponds to a dew-point temperature of —42 F.

Silica-gel is a hydrate of silicon dioxide which looks very much like rock salt. It is considered that the gel contains innumerable amicroscopic pores. It has been calculated[3] that the pores have diameters from 2.5 to 55 millionth microns. This being the case, it can be seen that a single granule of gel contains an enormous surface area on which vapors are adsorbed. Calculations[4] show that this internal surface area of 1 cu in. of gel is more than 43,000 sq ft. The water vapor which can be adsorbed by silica gel amounts to about 40 per cent of its weight. During this process of condensation, a heat of adsorption is given off,[5] which for the conditions of this plant is approximately 16 per cent greater than the heat of vaporization and is manifest in a temperature rise of the air being dehydrated. Since this gel is of the irreversible type, that is when heated it does not melt to a sol, heat may be used to again evaporate the adsorbed water.

Although the cycles involved are comparatively simple, the design of commercial equipment to accomplish these results involves many problems not apparent in a casual consideration. *First,* the process must be continuous. *Second,* an automatic control for this system is required to operate 13 dampers, start, control, and stop an oil-fired heater and a water-cooled heat exchanger, and start and stop one fan. *Third,* the infiltration and by-passing of moist air must be practically nil in order to obtain the required low humidities. This item is significant because the static pressures involved run as high as 8 in. of water.

The principal parts of the system discussed in this paper are: Two adsorbers, an activation air heater, a gel cooler, a control unit, and the necessary fans, duct work and dampers. The adsorbers are shown in Fig. 7. This plant is operated with six hours of adsorption, three hours for activation, and three hours for cooling. While one bed is on the adsorption cycle, the other one is on the activation-cooling cycles.

As far as possible a welded construction was used throughout the entire system. In a few cases, gasketed joints were used where exfiltration might occur, but they were never used where infiltration or by-passing could take place. The beds of gel are horizontal and are approximately 13 in. deep. The gel is supported on a corrosion resistant screen. This material is used so that the moisture condensed at the end of the activation cycle will not form rust and thereby close the openings. This construction is shown in section in Fig. 7. During adsorption the velocity of air through the plenum chamber above and below the gel is approximately 45 fpm while during activation it is approximately 20 fpm. The heat of adsorption and friction plus that transmitted through the walls gives from 19 F to 30 F temperature rise depending upon the time relative to the start of the adsorption period when the housing is still warm.

The dampers in the adsorbers and the others which are operated automatically are of the pneumatic type. These dampers are rectangular. The seats are machined from cast-iron sections. The gates are carried on the shafts of double-acting air cylinders. The shafts ride on pulleys on the ends opposite the cylinders. The pistons are equipped with leather packings or piston rings depending upon whether the damper is used for hot or cold air. A control unit actuates the pistons to open or close the dampers. On this unit an electric clock carrying a set of cams operates a mercury switch which in turn starts a small motor. Acting through two cams this motor operates two sliding valves which control the compressed air which moves the pneumatic dampers. One valve controls the dampers on the adsorbers while the other controls the dampers on the heater and gel cooler.

When No. 1 adsorber, for example, starts on adsorption, No. 2 adsorber starts on activation. Approximately 3,000 cfm of air are drawn from the equipment room for this purpose. After passing through a panel-type filter and the activation-cooling fan *G* of Fig. 3, it is forced through the oil-fired heater, through the connecting duct, to the gel bed, and out of the connection to the stack. The air makes four passes through the flat elongated tube sheets of the heater. The last pass is exposed to the highest temperatures. During the evaporation of moisture from the gel the effluent air temperature is essentially constant but as soon as the evaporation is completed, this temperature rises rapidly, showing a distinct *break.* The activation temperature is changed if the adsorbed moisture is varied either due to changing the volumes or the inlet humidity conditions. These temperatures are adjusted so that the *break* in the temperature of the effluent air occurs at the ends of the three-hour activation period.

The heater consists of four units. Oil is furnished

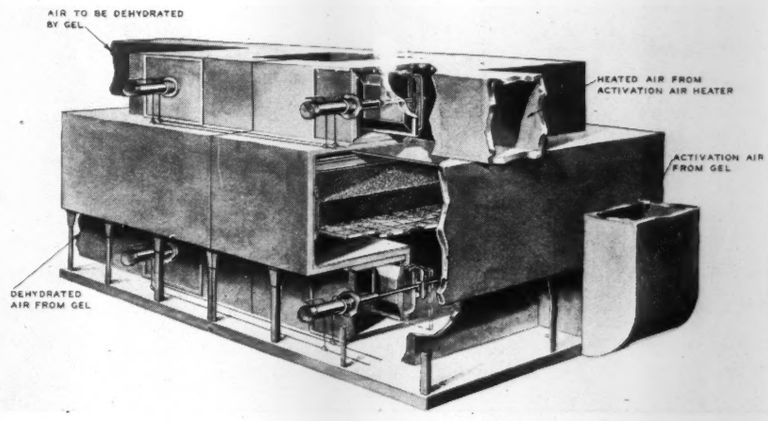


Fig. 7—The two silica-gel adsorbers showing pneumatic dampers, gel bed and bed supports

[3] Applied Colloidal Chemistry by Bancroft, p. 249.
[4] Story of Silica Gel, by J. C. Patterson.
[5] Patrick and Greider, *Journal of Physical Chemistry,* 9:1031-9 August, 1925.

from a supply tank to a reservoir before each unit by means of a circulating pump, the excess oil draining back from the reservoirs to the supply tank by gravity. Each unit has two burners of which one is fitted with a gas pilot. Oil is drawn from the reservoir and atomized by means of compressed air which is blown at an angle across the nozzles. The temperature control operates upon one burner of each unit. Thus, as the temperature exceeds the established value, it closes the air supply line to the control burners on each unit, which consequently go out. The products of combustion are removed by the induced draft fan *E* of Fig. 3 which discharges into the stack. The activating-cooling fan has been placed on the inlet side of the heater so that the heated-air side of the heat exchanger sections are under positive pressure; in this way no products of combustion can enter the gel beds through any cracks which may appear in these sections. The induced draft fan and the activating-cooling fan are equipped with centrifugal cut-outs so that the compressed air will be shut-off in case of failure of any of these fans, with the result that no oil is drawn into the heater. After the heater has operated for three hours it is automatically stopped, but the dampers of the air passages are not shifted until the interchanger sections have been cooled for 15 min.

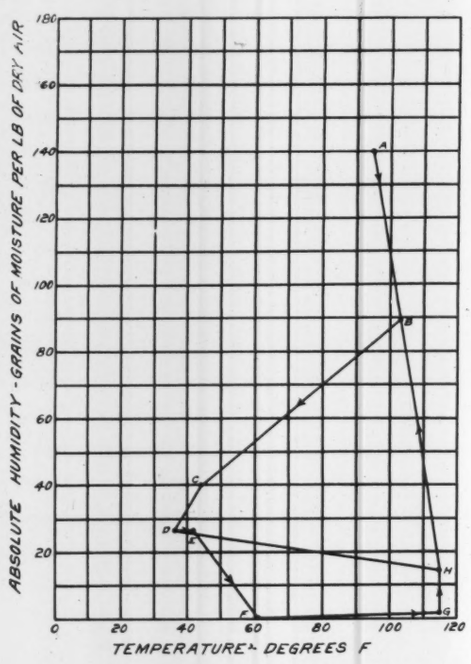


Fig. 8—Absolute humidity—temperature cycle for complete conditioning system

After the furnace cooling is terminated, the dampers are shifted and the gel cooler is connected to the adsorber which has recently been activated. In this cycle the air is circulated through the adsorber and extended-surface, water-cooled heat exchanger by means of the activation-cooling fan. The cooling cycle extends over a period of two hours and 45 min. during which time the temperature of the gel is lowered to 85 F. When No. 2 adsorber is cooled, it is automatically placed on adsorption and No. 1 adsorber is placed on activation.

Conditioned Storage Rooms

Dehydrated air is supplied to the toll storage room by a duct in the upper corner of the long side of the room opposite the vestibules. Diffuser inlets are placed at 5-ft intervals in the bottom side. The air outlets, which are located on the wall against which the vestibules are erected, are placed 4 ft above the floor. The air which is removed from the room by a recirculating duct to supply the loop storage room, is actuated by the higher static pressure in the toll storage room. However, a propeller type fan has been placed in the connecting duct to increase the rate of circulation if it becomes desirable to do so. The ducts in the toll storage room are shown in Fig. 2. The air removed from the toll room maintains the proper humidity in the loop storage room which is operated at a higher vapor pressure level and sometimes at a slightly higher temperature. As previously indicated, air is supplied to the latter room by the loop supply fan in the refrigeration unit when needed to maintain the proper temperature. The economy of recirculating some air from the loop room or using fresh air exclusively is dependent on outside conditions. Due to the sensible heat in the air, it was found to be cheaper to recirculate from the toll room to the loop rather than to the dehydration system.

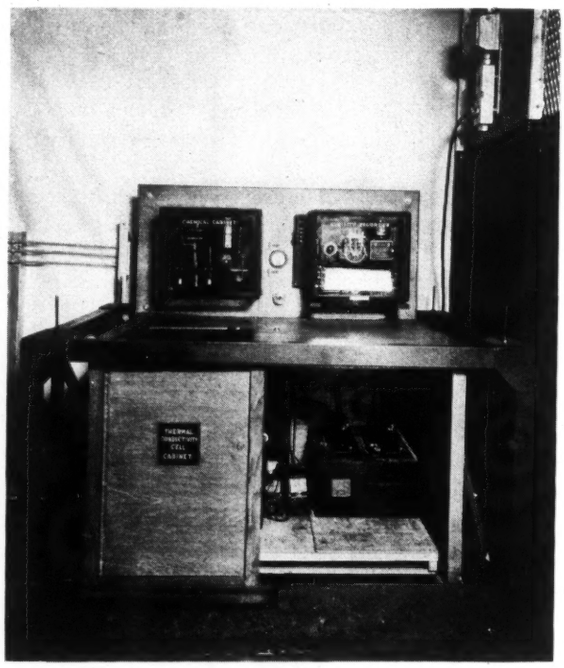


Fig. 9—Low absolute humidity recorder

Unit heaters are used to temper the air in each room. These were found to be more satisfactory than duct heaters because radiant heat did not affect the refrigeration or adsorption systems; there is no danger of freezing; repairs can be made easily; and during periods when the conditioning system is not in use the large fans need not be run, which results in a power saving.

The absolute humidity-temperature cycle for the complete conditioning system is shown in Fig. 8. Point *A*

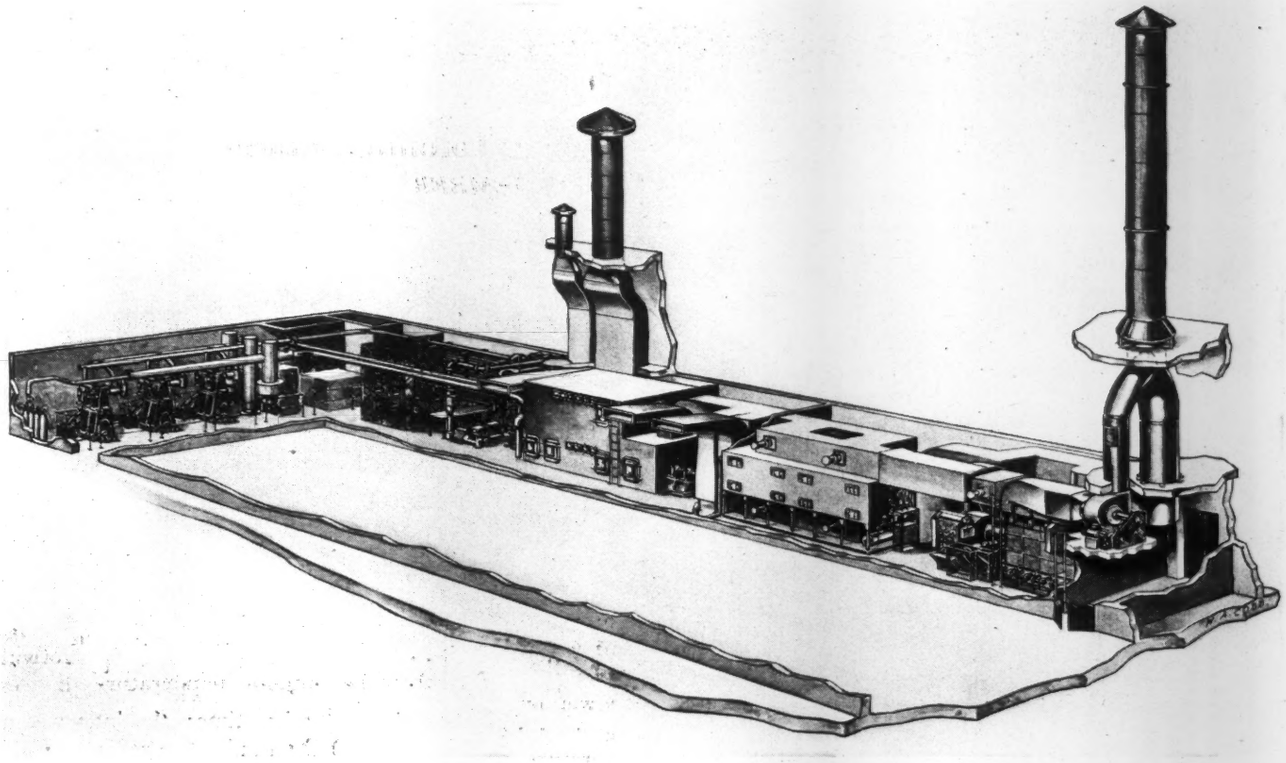


Fig. 10—Complete air-conditioning system for cable storage rooms showing compressors, humidity recorder, refrigeration unit, silica-gel adsorbers, activation air heater with stack, fresh air inlet and concentrator exhaust

represents fresh air at the maximum temperature and humidity. It is mixed with an approximately equal volume of recirculated air from the loop room to give the condition at *B*, namely 103 F and 89 grains of moisture per pound of air at the entrance to the water stage. Point *C* shows the condition at the end of the water stage and the normal condition at the end of brine stage is shown at *D*. As the air passes from the refrigeration unit through the two fans to the adsorption system, it is heated to the temperature at *E*. The silica-gel adsorbers dehydrate the air from *E* to *F* with a consequent temperature rise of 19 deg. The conditions in the toll storage room are shown at *G;* the relative humidity in this room is 0.5 per cent. As the air from the toll room is blown into the loop room which is at the same temperature there is a resulting rise in humidity to point *H*. The humidity in the latter room is 3 per cent. Air from the brine stage is delivered to the loop room along *DH* with a consequent rise in sensible heat and a decrease in latent heat. This cycle is based upon maximum conditions of operation; the average humidity conditions are somewhat lower. The average relative humidity in the toll room is 0.3 per cent and in the loop room 1.5 per cent.

The determination of such low humidities is an unusual problem. A continuously-recording absolute humidity [6, 7] instrument which is shown in Fig. 9 has been developed for this purpose. In this instrument, four platinum elements in suitable cells, all precisely alike, form the arms of a Wheatstone bridge. The containing cells are surrounded by an oil bath of constant temperature. A potential applied to the network causes appreciable currents in the arms, and if the heat dissipation is the same from all the elements the bridge remains balanced. In practice, the air to be tested is passed through suitable drying towers and returned to the other two arms. All the air is maintained at the same temperature by the oil bath. The thermal conductivity of air is dependent on its moisture content, and the elements in the drier air will develop a higher temperature than the other elements, thus increasing their resistance, and causing a voltage unbalance which is an index of the moisture content. From 0 to 10 per cent at 77 F it is sensitive to changes in relative humidity of 0.05 per cent which is 0.07 grains of moisture per pound of air.

The complete conditioning system is shown in Fig. 10. The compressors and traps are shown in the left hand side of the drawing. The dehumidifier is placed to the right of the condensers after which the adsorbers, activation filter and fan and activation air heater are installed. The first stack on the left is the brine concentrator exhaust, the one in the center is the fresh air inlet and the one at the right removes the products of combustion from the heater as well as the warm moist air which is discharged from the gel beds during activation.

In concluding, the author wishes to acknowledge the work done by other engineers on this project. B. J. Babbitt and C. R. Hoffman, particularly, have contributed many ideas used in the installation, as well as suggestions in the preparation of this paper.

[6] C. Z. Rosecrans, *Industrial and Engineering Chemistry* Anal. Ed. 2:129-134 April 15, 1930.

[7] A. C. Walker and E. J. Ernst, *Industrial and Engineering Chemistry* Anal. Ed. 2:134-8 April 15, 1930.

Some Observations on Heating Practice

By James Govan[1], Toronto, Ontario
NON-MEMBER

IN a paper[2] presented by the writer at the Semi-Annual Meeting of the Society, 1929, information was submitted to show that the data used by the Society for determining maximum heating plant requirements for buildings should be modified. The purpose of the present paper is to submit the results of certain observations which have been made since that time and to offer certain suggestions with respect to assumed outdoor temperatures used for heat loss calculations in order to reconcile if possible the apparent discrepancy between actual and estimated heating plant performances.

Research conducted by the Society[3] has shown that a large error may be introduced into the calculations by failure to consider the periodic character of the heat flow through the walls and roof of a building. As has already been pointed out,[4] the resistance to heat flow and the heat capacity of the structure combine to stamp out this wave amplitude. The observations made by the author in Canada would tend to substantiate this fact, particularly with respect to the *fly-wheel* effect of the resistance to heat flow. In other words, materials of high heat resistance produce time-lag effects in both summer and winter air conditioning systems which must be taken into consideration in calculating the size of such plants.

The architectural profession is much concerned with this problem in its efforts to provide the most comfortable and healthful conditions for the occupants of buildings at the lowest possible capital and annual operating costs. If it is practicable, as seems possible from studies of actual occupied buildings under severe climatic conditions, to maintain comfortable conditions in a type of structure requiring as much as 70 per cent less heating plant than would be the case for the same building built in the customary manner, then it is folly to continue to follow present methods.

Unheated Packing Shed

Figs. 1, 2 and 3 show the plan and section and exterior and interior views of a packing shed in Ontario having frame wall and roof construction, with a fireproof board lining on the inside, fireproof insulation fill in stud and joist spaces, fireproof board, waterproof felt and clapboard on exterior of studs, and roof boarding and built-up roofing on top of roof joists. The windows consist of two single-glazed casement sashes bolted together with thumb bolts. Large doors at each end of the shed are of the overhead opening type and are single thickness; the small door in the side wall is also of single thickness, and no calking or weather-stripping was used at either windows or doors. The building was designed and built to provide a low-cost shed that would maintain a fairly uniform temperature during the nursery stock shipping seasons of the spring and fall. During the winter months it is used for grafting, budding and other indoor nursery occupations.

Indoor and outdoor temperature charts were taken from October 27, 1930, to February 23, 1931, and from March 23 to 29, 1931. The outdoor bulb of the recording thermometer was on the east and least exposed side of the building, which is built on an open rolling piece of land well exposed to the wind. Notwithstanding the fact that the outdoor temperature dropped to zero or lower on 14 occasions during this period, and to a degree or two above zero at other times, the indoor temperature was as low as 30 F on only six occasions. The longest period of 30-F indoor temperature lasted only 24 hours and that followed a drop from 10 F to —17 F outdoors during the night of January 31-February 1.

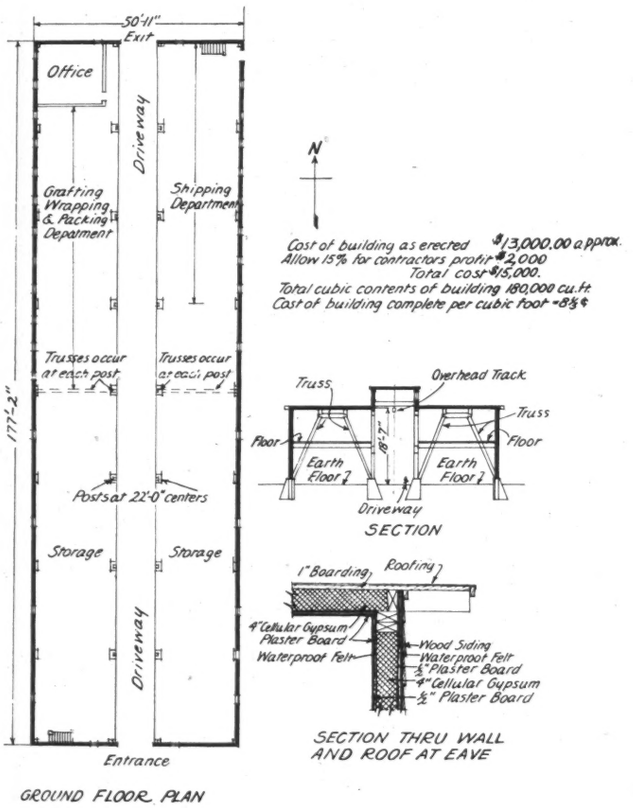


Fig. 1—Plan and section of packing shed without heating or cooling plant in which temperature records were kept for three years

[1] Member of firm of Govan and Ferguson, Architects and Associated Engineers.

[2] Time Lag as a Factor in Heating Engineering Practice, by James Govan (A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929).

[3] Coefficients of Heat Transfer as Measured Under Natural Weather Conditions, by F. C. Houghten and C. G. F. Zobel (A. S. H. V. E. TRANSACTIONS, Vol. 34, 1928).

[4] Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. M. Pugh and Paul McDermott (*Heating, Piping and Air Conditioning*, April 1932).

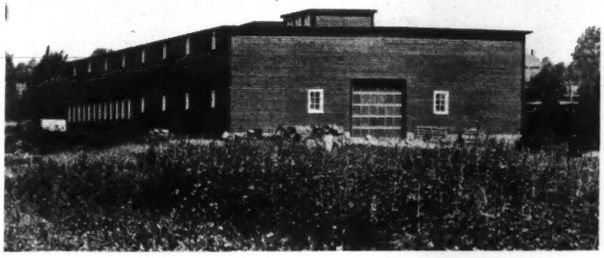


Fig. 2—Exterior view of packing shed

Fig. 3—Interior view of packing shed

1931. On January 28 it had dropped to —4 F, and during the night of the 30th-31st, to —4 F, rising to 10 F above at 3 p. m. on the 31st. The maximum variation indoors during that week was from 37 F to 30 F.

It is also worthy of note that, out of the 18 weeks, during 9 of them the outdoor temperature was consistently below the temperature indoors, and out of a ten-week period from November 24, 1930, to February 2, 1931, the outdoor temperature was higher than the indoor on only one occasion, namely, for about 8 hours on January 25, 1931, when the thermometer bulb hanging in the sun showed 40 F at about 2 p. m.

Suppose a heating plant were to be installed in this building to provide an inside temperature of 65 F for the men doing the winter grafting, etc. Disregarding the heat capacity of the structure, the outdoor temperature assumed would be —5 F or 15 deg below the lowest temperature on record, —20 F being the lowest recorded at the Nursery for 10 years. This would give a temperature difference of 70 deg when, as a matter of actual record, the inside temperature was maintained at 30 F or higher for two winters without any heating plant. If a temperature difference of only 35 deg were assumed (65—30), then the heating plant size would only be one-half that obtained by the usual methods of calculation. In other words, the plant ordinarily installed would be 100 per cent in excess of requirements.

The general formula used for calculating the amount of heating surface (radiation) required is as follows:

$$R = \frac{(H_v + H_i + H_t)\ (t - t_o)}{240} \qquad (1)$$

where

R = square feet of equivalent heating surface (radiation) required, based on an emission of 240 Btu per square foot per hour

H_v = heat required to warm air introduced for ventilation purposes from t_o to t, Btu per hour

H_i = infiltration heat losses, Btu per hour

H_t = transmission heat losses, Btu per hour

t = inside design temperature, degrees Fahrenheit

t_o = outside design temperature, degrees Fahrenheit

240 = heat emitted by one square foot of equivalent heating surface (radiation) Btu per hour

Formula 1 may, of course, be used for either uninsulated or insulated buildings, provided the proper value of H_t is used for the type of construction involved. This formula has been modified by the writer for calculating the amount of heating surface required when the walls and roof have a comparatively high heat resistance, as follows:

$$R = \frac{(H_v + H_i + H_t)\ (t - t_x)}{240} \qquad (2)$$

where

t_x = minimum indoor temperature that would be maintained indoors if no heating plant were provided (determined by observing the results in buildings after they are erected and modifying subsequent calculations accordingly)

Small Office Building

After the Semi-Annual Meeting 1929 an opportunity was sought to experiment with the small office building illustrated in Figs. 1, 1A and 2 of the paper presented at that meeting.[5] As then stated, the data regarding the heating surface in this office building had been checked by one of the then members of the A. S. H. V. E. Council, who satisfied himself that the amount of surface provided had not been over-estimated and was in accordance with the procedure set forth in THE GUIDE 1929.

Prior to the 1929-30 heating season, approximately one-third of each radiator was cut off and removed from the rooms. During that heating season it was found that the required temperature of the water at the boiler was still so low (in mild weather from 90 to 100 F) that the room temperature on the top floor

[5] Time Lag as a Factor in Heating Engineering Practice, by James Govan (A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929).

farthest away from the boiler dropped below 70 F. To avoid the necessity of raising the temperature of the water at the boiler to stimulate circulation, the flow on each of the two main feed lines was increased slightly by mechanical means. This corrected the difficulty and the plant was operated in that manner during the 1930-31 season.

Fig. 4 shows the boiler with circulating devices on the mains at *A*, recording thermometer bulb in end of boiler at *B*, boiler water recording thermometer at *C*, and part of closed tank equipment at *D*. Indoor temperatures were recorded in the front room on the first and second floors and in the accounting room on the ground floor at the rear of building, also outdoors on the southeast wall, using a bare bulb not shielded from the sun and protected from the effect of north and west winds by the building. It should be noted that there is a Dominion Government Weather Recording Station on the same street not more than 200 yards from this building, and whereas the chart at the building shows a low dip to 2 F on January 8, 1931, the Government Record taken away from any building shows —2 F. Similarly other comparisons at these two points are as follows:

	Building Chart	*Government Chart*
January 15, 1931	1	—3
January 28, 1931	4	—1
February 1, 1931	—4	—6
February 3, 1931	0	—4
February 5, 1931	10	—3
February 11, 1931	1	—3

The temperature of 10 F shown at the building on February 5, 1931, contrasted with the Government record of —3 F may be explained by the retained heat in the wall affecting the nearby thermometer bulb, and also the heat coming from open windows in the accounting room which were quite close to the bulb. Frequently this room was occupied by staff employees working at night. This would be aggravated if the windows on the southeast side of this room were being used for ventilation with a wind from the northwest.

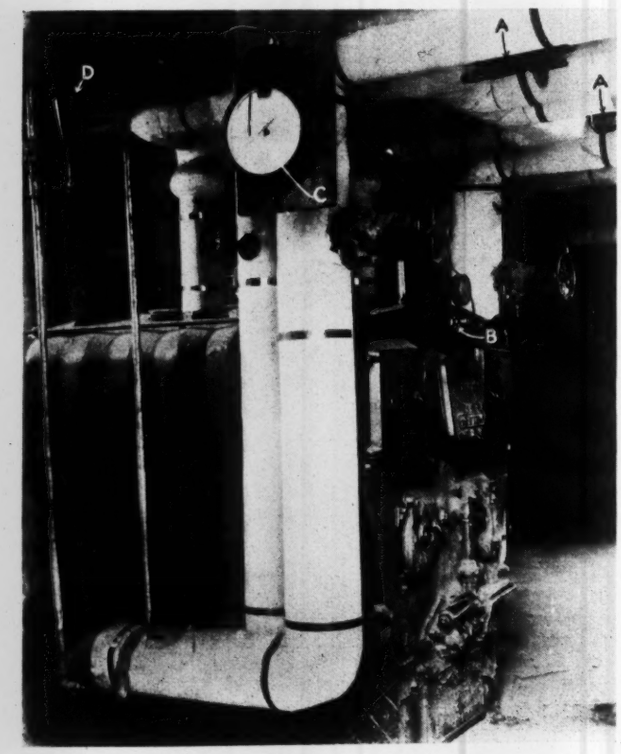


Fig. 4—Boiler in small office building

These charts show that from January 6, 1931, to February 11, 1931, the indoor temperature dropped below 70 F on only one occasion, when it reached 60 F. The outdoor temperature dropped below zero at the Government Station, which would correspond to the north and west sides of the building, on eight different days.

Table 1 shows the temperature of the water at the boiler for the period during which the charts were made. On February 3, 1931, and February 5, 1931, it was below zero outside and yet the boiler water was not raised above 96 F until the 7th, and then only to 143 F for about one hour, which was the highest temperature recorded at the boiler during the heating season 1930-31. Without the reduction of 33 per cent in heating surface the temperature of water at the boiler recommended by the manufacturers of the pressure control tanks used is 190 F at an outside temperature of 0 deg F.

Coal Consumption

As the coal used was not weighed exactly as used each season, the only accurate figure that can be certified is that 75 tons of buckwheat anthracite were used for the whole of three heating seasons 1928-29, 1929-30 and 1930-31. This gives an average consumption of 25 tons per season. As the building was only completed in the fall and early winter of 1928, the first season's coal consumption was naturally greater than that for the two following seasons, because of drying out of the building structure, etc. However, if 25 tons be taken as the yearly figure the facts regarding the poor combustion conditions at the boiler are interesting. It had been noted that due to the need for maintaing unusually low water temperatures at the boiler, it was practically impossible to maintain satisfactory fires, with the result that a very considerable proportion of the buckwheat sifted through the grates in an unburned condition. To check this the whole of the ashes for the season 1930-31 were kept in a pile and carefully sampled in the spring of 1931. This check gave over 20 per cent of the weight of ashes as unconsumed coal.

Further Reductions in Radiation for Season 1931-32

The studies made during 1930-31 suggested that the heating surface should be further reduced so that the boiler water temperature could be kept higher to maintain a better temperature head, and the grate area of the boiler reduced to obtain better combustion.

The total heating surface in use for the 1930-31 season was 1,296 sq ft. This was reduced by 448 1/3 sq ft, bringing the total installed down to 847 2/3 sq ft. Most of this reduction was made on the southeast and southwest sides of the building, because it had been noted that, notwithstanding previous reductions, the sun effect on these sides was quite pronounced. Fig. 5 shows the extent of the reductions in heating surface made since the original data were presented in the aforementioned paper.

The boiler is of the magazine-feed type with the gravity coal feed split two ways, so it was decided that

Table 1—Temperatures of Water at Boiler (See Fig. 4)[a]

Date, 1931	High	Low	Mean
Jan. 6	131	100	119.3
7	129	107	119.3
8	128	100	115.9
9	128	90	107.9
10	118	88	101.2
11	123	105	110.5
12	119	99	101.0
13	131	103	120.0
14	119	103	109.6
15	133	109	119.2
16	129	106	119.8
17	120	90	103.6
18	128	86	110.2
19	132	104	120.5
20	134	110	125.0
21	134	109	125.1
22	134	110	123.5
23	135	110	124.6
24	128	95	112.6
25	130	90	112.9
26	129	105	115.5
27	132	110	121.7
28	123	102	112.1
29	128	108	117.3
30	130	101	112.0
31	98	70	89.5
Feb. 1	97	75	90.3
2	...	...	.....
3	95	78	87.5
4	94	74	87.4
5	96	79	83.7
6	96	74	88.1
7	143	90	119.8
8	131	92	119.2
9	128	98	116.5
10	130	104	112.6
11	126	100	117.2
12	120	105	113.0
13	130	104	115.1
14	141	94	118.5
15	133	95	113.5
16	118	92	108.2
17	122	95	107.5

a. The mean temperature at the boiler was 111.1 F for the whole period, January 6 to February 17.

rather than cut down the length of each grate surface, one-half of the boiler should be shut down entirely. The boiler was operated in that condition for the whole of the 1931-32 season.

If the building in which the heating plant is installed had been of the ordinary uninsulated type of construction with the same kind of windows, the total heating surface required would be 2,830 sq ft. With only 847 2/3 sq ft in use, the reduction is 1,982 1/3 sq ft, which is approximately 70 per cent. An interesting fact is that the reduction in heating surface since this installation was previously reported to the Society in 1929 is approximately 50 per cent of the amount in use during the 1928-29 season, and as previously noted the plant at that time was designed in accordance with the procedure given in The Guide 1929, although the effect of heat capacity, discussed in The Guides 1931 and 1932, had not been taken into consideration.

Results During Season 1931-32

The reports obtained as to the comfort conditions in the building fully justify the reductions made in the fall of 1931. The weather was not as severe as in previous seasons, but the boiler water temperature records, in relation to the temperatures maintained in the building and the outdoor fluctuations, prove that there is ample capacity in the plant to take care of much lower dips in outdoor temperature than were experienced during the 1931-32 season.

Temperatures were recorded for the week of March 9 to 16, 1932. The indoor temperatures were taken in the accounting room at the rear of building ground floor, and on the first and second floors. On both the ground and first floors the temperatures were higher than was desired for comfort, the temperature on the ground floor being between 75 F and 80 F practically day and night, and on the first floor at 70 F or above.

These high temperatures were maintained because of conditions encountered in the large front room on the top floor. There the proportion of window to floor area is much larger than in any other part of the building. This room was originally planned for an assembly room for staff meetings, but has not been used for that purpose except on a few occasions. At other times it has been occupied by one or more individuals doing ordinary clerical work. This situation, coupled with the fact that the wood windows have no storm sash, and are in very loose condition, points to the need for increasing the relative heat retaining capacity of the room so that it will correspond with the other parts of the building. Alternatively, a small addition to the amount of heating surface in this one room would permit the maintenance of a lower water temperature at the boiler.

Table 2 gives the boiler water temperatures for this

Table 2—Boiler Water Temperatures for Week of March 9, 1932

9th Day	10th Day	11th Day	12th Day	14th Day	15th Day	16th Day
170	165	132	145	135	135	139
156	155	133	143	132	132	135
142	142	134	140	130	126	128
130	132	132	130	146	118	115
154	131	158	135	144	126	127
150	146	164	137	143	124	138
145	147	161	137	141	123	135
138	132	150	139	135	121	131
138	139	160	137	136	130	130
144	130	159	136	131	135	123
130	124	140	122	131	134	130
120	117	133	112	136	131	134
123	115	131	106	137	129	135
128	115	131	110	135	127	134
135	115	132	120	132	124	131
141	115	131	125	133	120	129
144	116	134	129	130	118	127
147	116	130	130	128	116	125
149	116	128	130	126	115	124
147	115	124	126	124	114	122
145	113	120	123	126	113	121
160	125	133	.....	122	132	120
168	130	148	.....	138	139	120
170	131	152	.....	130	141	129

same week and shows a maximum of 170 F and a minimum of 106 F, with an average of 133 F for the week. The average outdoor temperature for this period was about 25 F, the lowest being 4 F. According to the table of temperatures required for the closed system the boiler water temperature should have averaged 165 F for an average outdoor temperature of 25 F.

There were only two other occasions during the heating season when the boiler water temperature was over 170 F, *viz*, on March 7, 1932, when the chart indicated 173 F for about half an hour and 176 F for about one hour on the following day. The outdoor temperature on the 6th and 7th did not go above the freezing point even with the bulb in the sun at noon, and on the 8th a temperature of 8 F was recorded at 6 a. m.

The only conclusion that an architect can draw from a study of this kind is that it would be much better to spend the small amount of money necessary to make one comparatively insignificant room in a job equal in heat retaining capacity to all the other rooms by improving the windows, rather than to spend 100 per cent more than is necessary on the heating plant to have a margin of safety that can never be put to use for the whole job.

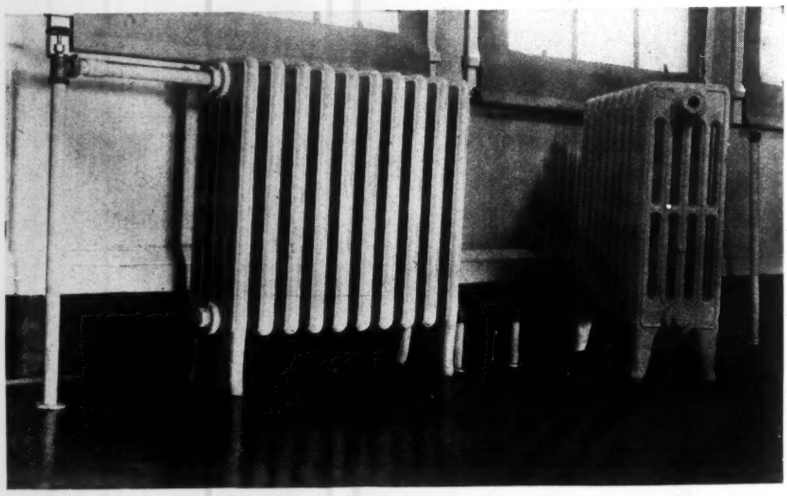
Bungalow in Northern Quebec

Many other examples can be cited, but Fig. 6 showing the plan of a bungalow residence in Northern Quebec will give some idea of what is being done in Canada's Northland. This building is of frame construction and the walls, floors and ceiling are insulated with a fireproof insulation fill. The roof is not insulated. No felt, calking or other weatherstripping was used around storm sash or regular sash or doors.

Steam was provided for this residence and other buildings at a pressure of 25 lb at the plant boiler room located about 1,000 ft from the residence. The 25-lb pressure was the highest used all winter, although a temperature of —38 F was recorded during the 1929-30 winter. As higher outside temperatures were experienced the steam pressure was reduced to 15 lb.

The engineer reported that the super-heating effect of the steam being reduced from the boiler pressure of 80 lb to 25 lb at the valve located at the boiler was offset by the drop in temperature in the length of line. Condensation was noted in this main steam line at the residence. This was part of the evidence cited for the belief that super-heating effect was eliminated.

All heating supply and return pipes connecting radiators were in the space between the floor and grade. Supply pipes were covered with 3 ply cell covering. Return pipes were not covered. This amount of heat took care of the plumbing, sewers, etc., carried under floor. Radiators were not trapped individually, but one



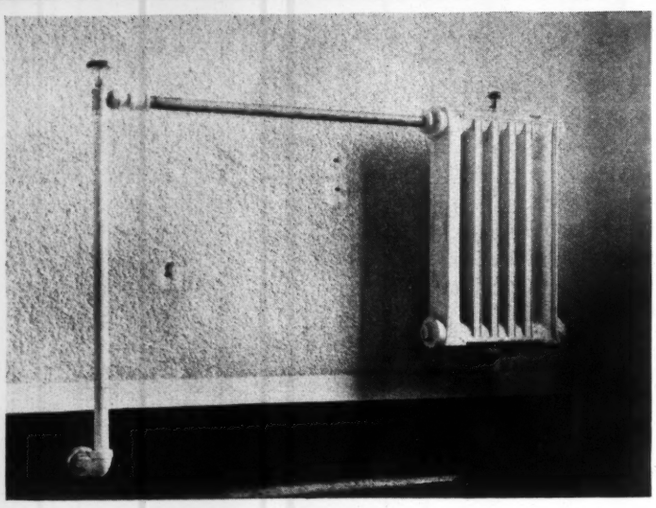


Fig. 5—Typical reductions made from original radiator installation[a]

[a] Operating results described in paper entitled Time Lag as a Factor in Heating Engineering Practice (see A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929). Cut made in summer 1929 removed one-third. Cut made prior to heating season 1931-32 shown by what is left connected up.

Above: Original installation, 26 sections or 91 sq ft; 7 sections or 24½ sq ft removed in 1929, leaving 66½ sq ft; 9 sections or 31½ sq ft removed in 1931, leaving 35 sq ft.

Below: Original installation, 3 sections or 27 sq ft; 1 section or 9 sq ft removed in 1929, leaving 18 sq ft; 1 section or 9 sq ft removed in 1931, leaving 9 sq ft.

trap was provided on the supply steam and one trap on the return. The condensed steam was not returned to the boiler plant owing to local conditions.

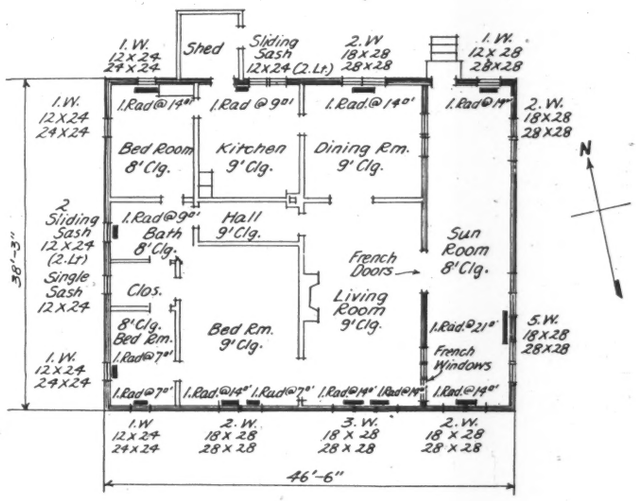
The sun room was used continuously throughout the winter just the same as any other room. Double doors into it from other rooms were open continuously. This room was used for reading and sewing and as a child's play room, etc.

On January 4, 1930, when the temperature outdoors was —12 F, only 67 sq ft of heating surface was in use, as follows:

Bath Room	9 sq ft
Large Bed Room	7 sq ft
Living Room	14 sq ft
Sun Room	28 sq ft
Kitchen	9 sq ft
Total	67 sq ft

On that date the following conditions were noted: large bedroom window open and storm sash open 3 in.; living room fireplace flue open, no fire in fireplace for 3 days; kitchen outside door open into closed back porch; French doors and windows open between sun room and living room.

Kitchen, hallway, bathroom, dining room, living room and sun room were open through; bulbs in bloom and



GROUND FLOOR PLAN

Total Heat Radiation Installed — 158 Sq. Ft.

4" insulation in walls shown blackened
3" insulation in floors
2 layers asphalt felt
4" insulation in ceiling

Fig. 6—Plan of bungalow subjected to temperatures below —40 F

ferns in all main rooms. Under these conditions the living room temperature was 70 F, with an outside temperature at —12 F and only 67 sq ft of heating surface connected. Steam pressure at valve in boiler room was 20 lb.

Two Hospitals Compared

Two typical hospitals, No. 1 in Ontario constructed in a manner somewhat similar to the small office building described in this paper, and No. 2 in Quebec with ordinary construction, furnished some interesting comparisons. The ratio of occupants and staff in the Ontario and Quebec buildings is approximately 1 to 3, and the services and accommodation provided in the Ontario building are, if anything, more elaborate than in Quebec, in proportion to the number of patients accommodated.

	Ontario Hospital No. 1	*Quebec Hospital No. 2*
Cubic space heated	160,000 cu ft	461,000 cu ft
Coal consumption per year	60 tons (Heating), 14 tons (Domestic water)	800 tons
Electric light and power cost per year	$785.65	$3,516.00
Gas cost per year for cooking	None	900.00

These two institutions are located in districts where the climatic conditions are similar and there is nothing radically different about them except the building construction and the mechanical and heating equipment. Sterilizing and other provisions affecting comparison just as complete in No. 1 as in No. 2. Both winter and summer comfort conditions for patients are better in No. 1 than in No. 2 and the capital cost per patient was at least 30 per cent lower in No. 1. Hospital No. 2 is not an abnormal example of such institutions. In fact, it may be said to be typical in both first and maintenance costs of the average institution of this kind built all over Canada and the northern section of the United States. The cost per 1,000 cu ft for heat, light and power compares very closely with several other typical institutions studied.

Summary and Conclusions

1. Under the extreme variations of temperature in Canada and the Northern States, the most important factor in determining heating plant size appears to be the outdoor temperature assumed, particularly when well-insulated types of construction are used.

2. Estimates of annual fuel consumption based on outdoor mean temperatures obtained from instruments shielded from the sun (as is the case at most of the Canadian Government Weather Stations) are likely to be seriously in error.

3. More attention should be paid to balancing the heat retaining capacity in all parts of a building, so as to avoid the necessity of providing a large, uneconomical reserve in heating plant capacity for only rare emergency use.

4. The heat radiating from the uncovered earth floor of the building in winter and absorbed by the earth in summer plays an important part in stabilizing the indoor temperature in the buildings studied by the writer. For instance, in the large storage shed illustrated in Figs. 1, 2 and 3, the earth temperature drops to about 40 F at a point 6 in. below the surface at the end of the winter.

5. The fact that an indoor temperature of 30 F or higher has been maintained in buildings without any artificial heating plants for three consecutive winters, where the outdoor temperature has dropped to —20 F, and also that 70 F has been recorded indoors without artificial cooling when the outdoor temperature was 90 F, indicates the need for further investigation to provide reliable data on the fly-wheel effect of different types of construction.

Corrosion as Related to Air-Conditioning Equipment

By R. M. Palmer*, New York, N. Y.
NON-MEMBER

IT IS customary in the air-conditioning industry to emphasize the factor of temperature and clarity of a given water and permit these properties to serve as a basis for adopting the use of any given available supply. Such a procedure is quite inadequate and has many times given rise to serious operating difficulties. A clear water having a satisfactorily low temperature throughout the year may prove to be an extremely dangerous water to use in air-conditioning equipment.

Properties of Water to Be Considered

What are the other properties of water which should be investigated before adopting a given water as a source of supply for condenser water and for spray chamber water for air conditioning systems? In general, a chemical analysis of the water should be made. It may not be necessary to make a complete chemical analysis, but it certainly is necessary to determine the total alkalinity, the total acidity, the total hardness and the *pH* values of a water before adopting it for use.

What will these analyses disclose? It is known that a highly alkaline water, if the alkalinity is due to calcium and magnesium carbonates, will be a scale-forming water. Scale may or may not develop in the cold. Calcium carbonate has a reverse solubility curve. That is, its solubility in water decreases as the temperature of the water increases. Because of this fact it is possible at times to use waters having a relatively high calcium carbonate content for spray washer purposes and for condenser cooling purposes. It must be appreciated, however, that scale may form at normal temperatures. The interior surface of a sample of piping taken from cold supply line of an office building in Buffalo, New York, had a deposit of the carbonates of calcium and magnesium. In this case the Niagara River supply, which flowed through this pipe, contains sufficient amounts of the carbonates of calcium and magnesium to bring about deposition of the scale in question, in the cold. The rate of scale deposition will depend not alone on the temperature factor but will also depend on velocity of flow.

Soft waters, or waters having a low calcium and magnesium carbonate alkalinity, are invariably corrosive. The corrosive action of such waters may be caused by the presence of dissolved oxygen, dissolved carbon dioxide or dissolved acids of organic or inorganic origin.

pH-Value and Hydrogen-ion Concentration

The determination of the *pH* of a water discloses certain information about what may be expected as to its corrosive properties. Pure distilled water will conduct an electric current to a very slight degree. This shows that a very small proportion of the water is dissociated into *H* and *OH* ions. By the mass law,

$$\frac{\text{Concentration of } H \text{ ions} \times \text{Concentration of } OH \text{ ions}}{\text{Concentration of Undissociated } H_2O} = \text{a constant} \quad (1)$$

Since the amount of undissociated water is relatively large, it can be taken as a constant, and Equation 1 therefore becomes

$$\text{Concentration of } H \text{ ions} \times \text{Concentration of } OH \text{ ions} = \text{a constant} \quad (2)$$

By electrical conductivity measurements, this constant has been found to be 1/100,000,000,000,000 or 10^{-14} at 22 C. Since in pure distilled water the number of *H* ions is equal to the number of *OH* ions, each must have a concentration of 1/10,000,000 or 10^{-7}.

This method of expressing *H*-ion concentration is very inconvenient and clumsy. Sorensen, therefore, suggested the use of the term *pH*, the *pH* value being the logarithm of the reciprocal of the hydrogen-ion concentration. Thus,

$$pH = \log \frac{1}{(H^+)} \quad (3)$$

Just as an acid solution is said to be normal when it contains 1 gram of ionizable hydrogen per liter, so a solution is said to be normal with respect to hydrogen ions when it contains 1 gram of ionized hydrogen per liter. Since $\frac{1}{(H^+)}$ is the reciprocal of the normality of *H* ions in a solution, the *pH* value can also be defined as the logarithm of the denominator expressing the normality of *H* ions such as *N*/10, *N*/100, *N*/1000, etc.

This method of expressing *H*-ion concentration has now been generally adopted. In the case of pure distilled water, in which the *H*-ion concentration is 0.0000001 or *N*/10,000,000, the *pH* value would therefore be the logarithm of 1/0.0000001 or of 10,000,000, which is 7.0. This value 7.0 is, therefore, the neutral point on the *pH* scale.

The *pH* values above 7 represent alkalinity and below 7, acidity. It does not follow, however, that a water with a *pH* above 7 will not be corrosive to both ferrous and non-ferrous metals. On the other hand, it does invariably follow that a water with a *pH* below 7 will be corrosive to both ferrous and non-ferrous metals. There have been instances where natural waters, having *pH* values as low as 4.5 were used regularly in air-condi-

*Vice president, Ferro-Nil Corporation.

tioning equipment but of course the spray washers of the air-conditioning equipment required replacement after a few months operation.

Typical Analysis

The following represents the analysis of samples of water before and after use in washing air:

	Before Use	After Use
Free carbon dioxide (ppm)	6	54.8
Total solids (ppm)	40	217
Calcium sulphate (ppm)	13	132
Hydrogen-ion concentration (pH value)	6.9	4.8

It will be readily recognized that in the case of the water which has been used for washing air a distinctly acid water with highly corrosive properties has resulted. The analysis of this water shows that its characteristics are entirely different from those of the water originally entering the spray chamber in question. What brought about this change? The water in question had merely been used to wash several hundred thousand cubic feet of air and this air, aside from containing a certain amount of insoluble matter, was just *plain air*. A study of the foregoing analysis discloses the fact that the water now carries a high sulphate content. How did this originate? The burning of sulphurous fuels results in the formation of sulphur oxides which in combination with moisture form the corresponding acids. These acids were absorbed by the water and account for the high sulphate content.

Water used for washing air may absorb relatively large amounts of carbon dioxide forming carbonic acid, a corrosive agent, thus accounting for the high CO_2 content of water used for washing air.

It will be readily seen from the foregoing that all water used for washing air absorbs acidic matter from the air washed. Therefore, proper steps should be taken to see to it that the water used does not become actively corrosive through chemical conditioning of the water.

Heat Transfer Efficiencies

In the case of all waters, whether they be corrosive or scale forming or both, the problem of lowered heat transfer rates will be encountered. Condenser coils or Baudelot coils coated with rust or scale or combinations of rust and scale mean materially lower rates of heat transfer with a consequent increase in operating costs. In this connection it was recently shown that the amount of rust accumulating on steel coils over a period of a few weeks was sufficient to lower the heat transfer rate of the steel in question to that of glass. Since the heat transfer rate of glass is very low as compared to steel it will be apparent that the heat transfer efficiencies of metals coated with products of corrosion or with scale are also relatively low.

Another factor in this problem of water supply which is becoming of increasing importance is that of regulatory measures which are now being introduced by various municipalities for the conservation of the water supply. This means the use of recirculated water on the condenser systems of air-conditioning equipment.

The schedules maintained for discarding recirculated water vary widely. The schedule followed in the case of two separate air-conditioning units recently inspected, which were located in the same building, will serve as an illustration. Both of these systems were operating on recirculated water for spray washers and for condensers. In one of the systems it was the practice to discard all recirculated water weekly and wash out any insoluble matter which might have collected during the operating period. In the case of the other system no regular schedule of discarding the recirculated water was carried out. At the time inspection was made, the water being used had been recirculated for a period of 41 days in one of the systems and for five days in the other system. The virgin water used had a total dissolved solids content of 60 ppm; after five days' use the total dissolved solids content had increased to 110 ppm; after 41 days' use the total dissolved solids content had increased to 285 ppm.

It is obvious that such radical changes in the analysis of the water used will bring about corresponding changes in the corrosive or scale-forming properties of the water. Actually it was found in the case of the system where no regular schedule for discarding water was used that the efficiencies of the system were lower than those obtained in the system where schedule of discarding water weekly and cleaning out system had been followed.

Location and Design of Air Intakes

Another interesting factor which is often of prime importance in this problem is the location and design of air intake ducts. It is natural for the architect to call for the location of such ducts at points which will conform to the architectural scheme developed. This will often result in locating ducts at points where excessively polluted air will be used. For example, ducts have been located at points close to smoke stacks and prevailing winds have at times blown stack gases into air intake ducts.

An unusual source of pollution was recently discovered. An air intake duct had been placed a few feet away from an exhaust duct leading from a storage battery room. The air from the storage battery room contained certain amounts of sulphuric acid mist which was sucked into the air intake duct leading to the air-conditioning equipment. Such a condition, unless immediately remedied, would lead to rapid deterioration of air-conditioning equipment. In the case in question, remedial measures were taken to condition the water so as to neutralize the acid absorbed.

Conditioned Water for Correcting Corrosion and Scale Formation

In the preceding discussion the statement has been made that chemical conditioning of water would alleviate troubles caused by corrosion and by scale formation. The question may now be logically asked, How can water be chemically conditioned to alleviate corrosion and scale formation?

The corrosive action of certain municipal water sup-

plies has been the cause of concern for many years. Ten years ago an effort was made to reduce the corrosive action of the water supply of Baltimore, Maryland, by chemical conditioning. The virgin water of this city is very soft having a low total alkalinity and a *pH* value of 6.8. The water was treated with lime, thus increasing the total alkalinity and raising the *pH* to 8.5. It was found that such a treatment brought about the deposition of a thin film of calcium carbonate on piping through which the water flowed and that this film acted as a protective medium against the corrosive action of the water. This method of treatment is still in use in Baltimore and has been adopted by other municipalities. The treatment is open to this objection, it increases the soap hardness of the water. The reason for citing this treatment has been to set forth the possibilities of chemically conditioning a water so as to make it non-corrosive.

The chemist is not confined to the use of lime alone for such purposes. Proper control of the total alkalinity and *pH* of a water may be obtained by the use of various alkaline substances. In general, *pH* values should be maintained at levels between 8.0 and 9.0 and the alkalinity should be maintained at levels which will result in the formation of a protective film and at the same time prevent building up of a film to the point where heat transfer rates will be adversely affected. The problem must be handled so as to prevent either over-treatment or under-treatment of the water. The mere dumping of an alkaline material into the water will prove not only inadequate, but definitely unsatisfactory and may give rise to difficulties more serious than those it is called upon to remedy. On the other hand, a properly regulated treatment with regular analysis both of treated and untreated water will afford optimum protection both from the standpoint of scale formation and corrosion.

The wide variety of characteristics exhibited by the various waters used in air-conditioning equipment does not permit going into detail in connection with the problem of chemical conditioning. Each water adopted for use must be analyzed. In addition, the same water must be analyzed after use and changes in its composition noted. Such a study will make possible the prescribing of proper treatment for the particular conditions encountered. Generalities are always dangerous. This is especially true in the case of water treatment problems.

Users of air-conditioning equipment are prone to criticize the manufacturer of the equipment for troubles which may arise and to hold the design of the equipment as being responsible. It is the writer's belief that, taking into consideration the advanced stage of development of air-conditioning information, the well established and reputable manufacturers of air-conditioning equipment are seldom responsible for difficulties encountered, such as rapid depreciation of equipment, caused by corrosion. If the operators of air-conditioning equipment will give some attention to the matter of water analysis and water conditioning, many of the troubles now encountered will be solved.

It should be pointed out that corrosion problems are not confined to those parts of air-conditioning equipment in which water is used. Corrosion takes place in air ducts, fans, eliminator plates and other parts of air-conditioning equipment which are used primarily for transfer of air. For example, gases such as hydrogen sulphide, which are some times encountered in the rayon industry, have given rise to corrosion problems of air transfer equipment.

Summary and Conclusions

This paper discusses the problem of water supply as related to air conditioning equipment. It is pointed out that characteristics of waters used vary widely. Certain waters are corrosive, other waters are scale forming. Waters often bring about high rates of depreciation of apparatus and lower operating efficiencies. It is shown that chemical analysis of waters followed by chemical conditioning will obviate many of the difficulties encountered.

Research Contributors

By G. L. LARSON,
Chairman Committee on Research

THE Committee on Research is greatly encouraged and inspired by the generous cooperation of the profession and industry in supporting the 1932 Research Program of the Society. The work being carried on is of fundamental character, and will supply authoritative data on a variety of subjects of which little is known at present. It is gratifying to know of the esteem in which the Research work is held by both members and others, and a striking evidence came to the Committee in a letter recently received which said:

"We certainly want to keep our shoulder to the wheel, but 'balancing the budget' is as much of a problem for the manufacturer as it has proved to be for the Government. Distasteful as the omission is, the budget committee's program for this strenuous year omitted all subscriptions.

"Every rule, they say, has its exception. I am glad to say that it proves true in this case. It is a pleasure to enclose check, which, though less than both you and we would wish, is a token of good will toward and support of the work of the Laboratory."

Expressions of this kind are heartening to the Committee on Research and make it easier to carry on.

Those who have placed their names on the Honor Roll in support of the 1932 Research Program are listed below:

American Oil Burner Association
Copper and Brass Research Association
Heating and Piping Contractors National Association
National Association of Fan Manufacturers
National Association of Ice Industries
Ventilating Contractors Employers Association
Aerofin Corporation
American Air Filter Company
American Blower Corporation
American Radiator Company
Barber-Colman Company
Barnes & Jones
Buckeye Blower Co.
A. M. Byers Company
Buffalo Forge Company
Bush Manufacturing Co.
W. H. Carrier
Carrier Engineering Corporation
Celotex Company
Clarage Fan Company
Crane Company
Detroit Edison Company
Detroit Stoker Company
C. A. Dunham Co.
Frigidaire Corporation
Garden City Fan Company
General Electric Company
Hoffman Specialty Company
Holtzer-Cabot Electric Co.
Howard Iron Works and Alberger Heater Company
Ilg Electric Ventilating Co.
Illinois Engineering Company
Johns-Manville Corporation
Johnson Service Company
Leeds & Northrup Company
National Regulator Company
Herman Nelson Corporation
J. J. Nesbitt, Inc.
New York Blower Corporation
Owens-Illinois Glass Company
Peerless Unit Ventilation, Inc.
Petroleum Heat & Power Company
Powers Regulator Company
B. F. Sturtevant Company
Trane Company
United Engineers and Constructors
Warren Webster & Company
Westinghouse Electric & Manufacturing Company
Youngstown Sheet and Tube Company

NOMINATIONS FOR 1933

The Nominating Committee appointed to select candidates for Officers of the Society for the coming year, 1933, submits the following list of nominees:

For President:

W. T. JONES, Boston, Mass.

For First Vice-President:

C. V. HAYNES, Philadelphia, Pa.

For Second Vice-President:

JOHN HOWATT, Chicago, Ill.

For Treasurer:

D. S. BOYDEN, Boston, Mass.

For Members of the Council:

Three-Year Term

R. H. CARPENTER, New York, N. Y.
J. D. CASSELL, Philadelphia, Pa.
F. C. MCINTOSH, Pittsburgh, Pa.
L. WALTER MOON, St. Louis, Mo.

Respectfully submitted,

NOMINATING COMMITTEE

H. M. HART, *Chairman*

Chapters	*Representative*
Cleveland	C. F. EVELETH
Illinois	H. M. HART
Kansas City	B. NATKIN
Massachusetts	C. P. YAGLOU
Michigan	W. G. BOALES
Western Michigan	S. H. DOWNS
Minnesota	C. E. LEWIS
New York	H. M. WYLIE
Western New York	JOSEPH DAVIS
Ontario	W. G. SHEPPARD
Pacific Northwest	——
Philadelphia	W. ROY EICHBERG
Pittsburgh	F. C. MCINTOSH
St. Louis	C. A. PICKETT
Southern California	L. H. POLDERMAN
Wisconsin	J. G. SHODRON

In accordance with the provisions of the Amended Constitution, as given, ballots containing the names of the above candidates will soon be sent to the membership for voting upon prior to the Annual Meeting in January.

Art. 9—*Section* 2. The Nominating Committee shall consist of a member designated by each Chapter, which shall effect its own organization and elect its own Chairman.

Section 3. The Committee may meet at the call of the Chairman, for discussion, but the official meeting shall be held at the Semi-Annual Meeting of the Society, at which time the candidates for each of the offices of the Society to be filled at the next Annual Meeting shall be selected by a majority vote of the members of the Committee present and voting. Five shall constitute a quorum.

Section 4. The Committee shall first secure the consent of all candidates selected, and shall, if possible, announce the names of the candidates at the Semi-Annual Meeting, but in any case the names of the candidates shall be certified to the Secretary of the Society at least four months before the next Annual Meeting. The Secretary shall publish these names in the October issue of THE JOURNAL.

Art. 10.—*Section* 3. The Secretary shall prepare ballots with the names of all candidates and forward them to the members at least thirty (30) days before the date of the Annual Meeting.

Nominations for Members of Committee on Research

Although committees of the Society are usually appointed, in view of the great importance of the Committee on Research and the financial responsibility it would be called upon to assume, it has been made more representative of the entire membership of the Society by a process of *election*. The election is governed by the By-Laws for the election of officers, with the single exception, that Members of the Committee on Research are nominated by the Council instead of by a Nominating Committee.

In accordance with the Regulations for the Government of the Research Laboratory, adopted at the January, 1919, Meeting, and revised January, 1930, the Council announces the nomination of the following members of the Committee for election to succeed those members whose present terms expire January, 1933:

3 Year Term

ALBERT BUENGER, St. Paul, Minn.
S. H. DOWNS, Kalamazoo, Mich.
H. N. KITCHELL, Cincinnati, Ohio
HOMER R. LINN, Chicago, Ill.
PERRY WEST, Newark, N. J.

1 Year Term

E. N. SANBERN, Philadelphia, Pa.

The regulations governing the nomination and election of members of the Committee on Research are as follows:

SECTION I—ORGANIZATION

Research Committee

1. There shall be a standing committee known as the Research Committee, consisting of fifteen members, each serving for three years, and five retiring each year.

a. The Council shall nominate previous to October first of each year five members to fill the vacancies of those retiring at the next Annual Meeting.

b. The nominations made by the Council shall be published in the October issue of the Society's JOURNAL.

c. Prior to December first of any year, any ten members, over their own signatures, may nominate one or more additional members of the Research Committee, and such additional nominations shall be placed on the ballot opposite the nominations made by the Council.

d. The election shall otherwise conform to the regulations provided for the election of officers of the Society.

e. Vacancies may be filled by the Council, such persons chosen by the Council to serve until a successor is elected at the next Annual Meeting.

Chamber of Commerce Building

Cincinnati to Entertain A. S. H. V. E. at 39th Annual Meeting

THE 39th Annual Meeting of the Society is to be held under the auspices of the Cincinnati Chapter at Hotel Gibson, January 23 to 25, 1933.

A three-day program of business, technical papers and entertainment is being provided for the enjoyment of the members and their guests who will assemble when Pres. F. B. Rowley calls the meeting to order on Monday, January 23, at 9:30 a. m. Morning and afternoon sessions are to be held Monday and Tuesday and the meeting will adjourn at noon on Wednesday. Among the papers submitted for discussion are:

Study of the Application of Thermocouples to the Measurement of Wall Surface Temperatures, by A. P. Kratz and E. L. Broderick.

Air Infiltration through Steel Framed Windows, by D. O. Rusk, V. H. Cherry and L. Boelter.

Corrosion in Air Conditioning Equipment and its Prevention, by R. M. Palmer.

A varied program is planned and other subjects to be discussed are: Refrigeration as Applied to Air Conditioning, a Study of Residence Cooling in Summer, Oil Burner Performance, Heat Dissipation from Insulated Hot Water Piping and Economics of Smoke Abatement.

From the Research Laboratory of the Society in Pittsburgh, three of the promised papers are:

Effect of Cold Walls on the Desired Effective Temperature for Comfort.

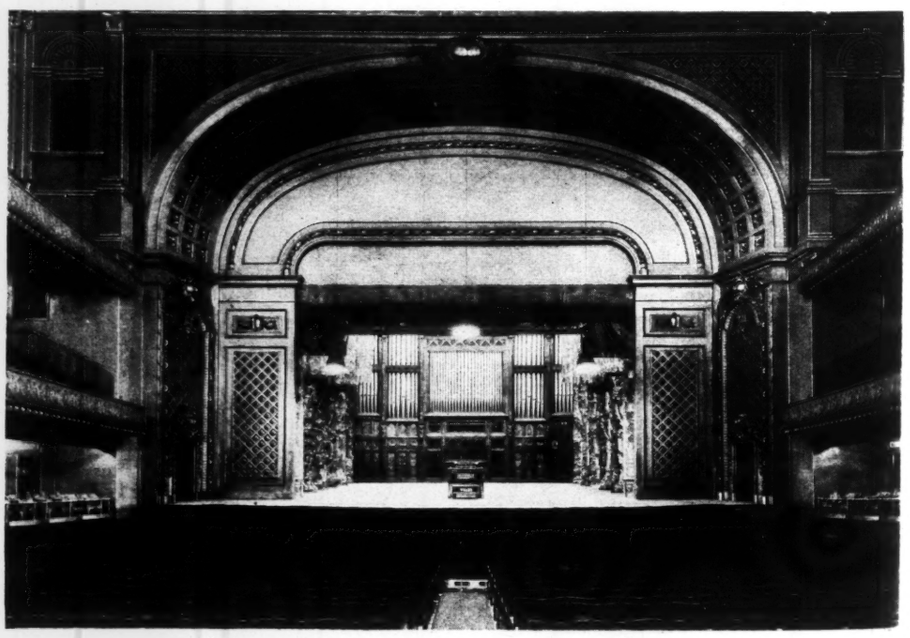
Capacities of 1½-in. and 2-in. Return Risers and Mains Carrying Water and Air in Different Percentages.

Rate at Which the Return Side of a Practical Steam Heating System Returns Air and Condensate During the Heating-Up Period.

The entertainment program is rapidly developing under the guidance of the General Chairman, W. C. Green, Cincinnati, and committees have already been appointed to handle Reception, Registration, Banquet, Ladies' Entertainment, Transportation, Publicity and Finance.

The meeting headquarters, Hotel Gibson, is conveniently located in the downtown business and amusement district, convenient to transportation terminals. Special attention will be paid to the ladies' program as it is the committee's desire to make their three-day stay a most enjoyable one.

Those members of the Committee on Arrangements who have



Music Hall Auditorium

accepted the responsibility for handling the innumerable details of the meeting are:

W. C. GREEN, *General Chairman.*
H. N. KITCHELL, *Vice-Chairman.*
J. J. LASALVIA, *Chairman,* Reception Committee.
E. B. ROYER, *Chairman,* Registration Committee.
I. B. HELBURN, *Chairman,* Entertainment Committee.
H. E. SPROULL, *Chairman,* Banquet Committee.
R. B. BRENEMAN, *Chairman,* Ladies' Committee.
R. W. SIGMUND, *Chairman,* Transportation Committee.
J. J. BRAUN, *Chairman,* Publicity Committee.
C. J. KIEFER, *Chairman,* Finance Committee.

Cincinnati is located at a central point and a large attendance of the membership is expected for this meeting in January. Special railroad fares at reduced rates for round-trip will be in effect and hotel rates are moderate.

Members of the Society will receive adequate notice of the meeting through literature provided by the Cincinnati Chamber of Commerce and through special notices from the headquarters, which will transmit hotel reservation cards and railroad certificates.

The Cincinnati members are keenly interested in having a large number of their fellow members to entertain during the last week in January.

Steam Power Plant Engineering

The major problems involved in the design of power plant apparatus are covered in a new book prepared for steam power engineers by L. A. Harding, Buffalo, New York, an outstanding authority in the profession. The book contains 13 chapters commencing with a discussion of fuels and combustion, and follows with treatment of boilers, furnaces, stokers, pulverizers, oil burners, superheaters, desuperheaters, resuperheaters, air preheaters, feedwater heaters, deaerating heaters, evaporators, water purifiers, pumps, steam engines, turbines, regenerators, reheaters, condensers, cooling towers, pipes, fittings, valves, heat coverings, accessories, etc.

There are 777 pages of data in this cloth-bound volume, which is 6x9 in. in size and is published by John Wiley & Sons, Inc., New York.

Air Conditioning for Comfort

A book of 244 pages has been prepared by S. R. Lewis, Consulting Engineer, and published by Engineering Publications Inc., Chicago, Ill. Twelve chapters are required to describe the fundamentals which govern heating and cooling for human comfort, the practical formulas for computing the component parts of heating and cooling systems and actual design data on dwellings and restaurants. A convenient table of contents precedes each chapter and aids in the rapid locating of desired information.

Generous use has been made of data from the A. S. H. V. E. GUIDE, the Research Reports of the Society and other authoritative sources. The book is written in engineering terms and engineers will find it a valuable source of information for reference. The book is bound in brown cloth, is 5½ x 8½ in. in size, stamped in gold and is adequately cross-indexed.

Cooling and Air Conditioning for Comfort

Cooling and Air Conditioning for Comfort—Theory and Calculations is the name of a paper-covered 96-page booklet prepared by William Goodman, which also serves as a section of *A. C. E.* Handbook, recently issued.

The author gives fundamental principles and develops a complete theory and set of formulas for various problems which rise in air conditioning practice. The book contains a large number of examples, many being worked out in detail and the various tables and curves will be found helpful to the reader interested in air conditioning problems. The publisher is the Aerologist Publishing Co., Chicago, Ill., which also issues the *A. C. E.* Handbook bound in looseleaf form, prepared by some well-known engineers in the heating and ventilating profession.

Death of H. F. Bray

Word has been received of the death of Harry Franklin Bray on September 19, at Barnes Hospital, St. Louis, Mo.

Mr. Bray was born September 25, 1868, at Chicago, Ill., and from 1890 to 1905 served as engineer at various large power plants in that city. From 1905 to 1921 Mr. Bray was active as a contracting erecting engineer in the Southern Illinois coal district. His career as a heating engineer began in 1921 and at the time of his death he was sales representative in Central and Southern Illinois for the Excelsior Stove & Manufacturing Co., located at Quincy, Ill.

Mr. Bray was elected a Member of the Society in June, 1932, and was highly respected by his many friends and acquaintances.

For many years Mr. Bray had lived in West Frankfort and the Officers and Council wish to extend their sincere sympathy to his daughter and grandchildren, who survive and with whom he made his home.

Death of J. Stuart Knee

The death of J. Stuart Knee, of Grand Rapids, Mich., occurred on September 26, as the result of injuries received from being struck by an automobile on September 13.

Mr. Knee was elected a Member of the Society in October, 1931, and his interest in and enthusiasm for the organization were always in evidence.

Born at Greenville, Mich., on November 18, 1876, Mr. Knee had had 36 years of experience in the heating, ventilating and plumbing field and at the time of his death was manager of the heating department of the Heat Equipment Corp.

The Officers and Council join in expressing their sorrow at the sudden passing of their fellow member and extend their sincere sympathy to his widow and family who survive.

Special Meeting of the A. S. H. V. E.

All members of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS have received notice by mail of a special meeting of the Society to be held Monday, November 21, at 5:30 p. m., in the Building Trades Club, 2 Park Ave., New York City, for the purpose of acting upon a resolution sponsored by the Council, which would amend the provisions of Article III of the By-Laws, relating to Initiation Fees and Dues, so that setting of fees for the years 1933 and 1934 would be delegated to the Council.

This special meeting will be followed by the regular November Meeting of the New York Chapter and all in attendance are cordially invited to participate. A Symposium on Ventilation is to be held and the featured speakers are William H. Driscoll, New York, Chairman, Committee on Ventilation Standards; A. J. Nesbitt, Philadelphia, Pa., and Prof. C.-E. A. Winslow, New Haven, Conn.

Local Chapter Reports

Kansas City

September 13, 1932. Pres. W. A. Russell called the meeting to order at the Hotel Ambassador and after receiving various committee reports the proposed meeting of Kansas City and St. Louis Chapters at the University of Illinois in Urbana, October 15, was discussed in detail with the result that 12 of the members present pledged themselves to attend. It was suggested that the Secretary prepare a list of all Chapter members who could go in order that necessary reservations could be made at Urbana.

Prof. G. L. Larson, University of Wisconsin and Chairman of the Society's Committee on Research, was introduced and spoke on Some Things We Do Not Know About Heating and Ventilating. He described in a most interesting manner some tests made at the University of Wisconsin showing the facts of temperature and humidity in rooms during periods of maximum occupancy. These studies made at student functions showed, for instance, that during a dance the ballroom was pre-cooled to 60 deg and after a brief period of dancing air introduced at 22 deg did not reduce the air temperature to 70.

E. K. Campbell, member of the Council, stated that President Rowley would be in Kansas City on October 24 and it was unanimously voted that the regular October Meeting of the Chapter be held on that date in order that the President might visit the Chapter.

Secretary D. D. Zink presented the report of the Educational Committee which outlined plans which would provide a 30 min presentation at its Chapter Meetings on some phase of heating and ventilation. The first subject on this program was Air Conditioning and H. D. Betz, a member of the Kansas City Chapter, gave an able presentation of the condition of the industry as it is today and with the aid of slides showed the trend of future developments.

The program of the evening concluded with a motion picture presentation on Insulation and Asbestos Products.

Minnesota

September 12, 1932. The first meeting of the 1932-33 season was held in the Banquet Room of the Minneapolis Y. M. C. A. at 6:00 p. m., with W. F. Uhl, newly elected president, presiding and 50 members in attendance.

The season was started a month earlier than usual and Prof. G. L. Larson, University of Wisconsin, Chairman of the Committee on Research of the National Society, was our guest speaker for the evening.

Following the dinner, guests were introduced and the regular business meeting took place. The minutes of the last meeting were read and received their approval. Various committee assignments for the year were announced by President Uhl, and the new committee chairmen were called on for reports of work to date. The Program Committee, headed by A. B. Algren, outlined the tentative set-up. M. H. Bjerken, Chairman of the Publicity Committee, was congratulated upon the announcements secured covering Professor Larson's visit. Mr. Bjerken outlined plans of his committee for obtaining publicity on coming meetings. S. A. Challman gave a report on the Society's Semi-Annual Meeting at Milwaukee.

Upon completion of the regular business, Mr. Algren introduced the National President, F. B. Rowley, who is also a member of the Minnesota Chapter, who in turn presented Prof. G. L. Larson as the speaker.

Professor Larson chose as his subject, Some Things We Don't Know About Heating and Ventilating. He outlined the necessity of continuing the Society's research work, and cited 4 items needing consideration, such as: (1) Design of heating for high office buildings with the determination of where filtration stops, the neutral zone and where ex-filtration begins. (2) The design of heating and ventilating equipment for high rooms such as field houses. (3) The design of heating and ventilating systems for auditoriums used for many purposes. (4) The ventilation of school rooms with special regard to exhaust ventilating ducts. Professor Larson cited the above 4 problems as being only a few of the unknowns requiring additional research in order that the designing engineer may have sufficient data to insure the proper handling of his problem.

At the conclusion of Professor Larson's talk, questions were asked and answered, and the Minnesota Chapter was very grateful to Professor Larson for his message.

St. Louis

October 5, 1932. The regular October meeting of the St. Louis Chapter was held at the American Hotel, following a luncheon.

Pres. J. M. Foster called the meeting to order and the minutes of the previous session were read and approved.

The proposed Joint Meeting of the Kansas City, Illinois, Wisconsin and St. Louis Chapters, scheduled to be held at the University of Illinois on October 15, was discussed and about 12 members of the St. Louis Chapter announced their intention of being present.

President Foster announced that the Nominating Committee must be chosen and called for nominations. Messrs. L. W. Moon, J. W. Cooper and E. A. White were unanimously selected and in addition, the chair appointed G. W. F. Myers and G. A. Helwig as members. The committee was instructed to present a slate to the Secretary of the Chapter at least 30 days prior to the November Meeting.

As there was no further business, the meeting was adjourned.

September 14, 1932. The opening meeting of the fall season was held at David Ranken Trade School through the courtesy of Superintendent Turner. Reports of the Officers and Committees were requested by Pres. J. M. Foster and Treasurer Walters announced a good balance in the Treasury and Paul Sodemann, Chairman of the Program Committee, announced that a joint meeting with the Kansas City Chapter would be held October 15 at the University of Illinois, Urbana. Secretary Davis was requested to communicate with Chapter members and prepare a list of those who were able to attend.

Prof. G. L. Larson of the University of Wisconsin and Chairman of the Committee on Research, was introduced and gave a constructive talk on investigations of air infiltration and stratification under the title Some Things that We Do Not Know About Heating and Ventilating. The paper indicated the necessity for further research on problems about which little or no information is available. At the conclusion of the meeting under the direction of Messrs. Turner and Rodenheiser the members and guests present made an inspection of the Laboratories of the David Ranken Trade School.

At the business meeting, it was decided to have the regular October Meeting start with a noon day luncheon, in order that a business session might be held prior to making the trip to the University of Illinois. It was decided that the November Meeting would take the form of a social function in view of the very excellent spring party which was enjoyed by the Chapter members and their ladies.

President Foster paid tribute to the work of the Code Committee of which L. Walter Moon is Chairman, who had worked hard and earnestly in the preparation of a heating and ventilating

code for the city of St. Louis in cooperation with the Joint Engineering Council.

The meeting was attended by 13 members and guests among whom was H. W. Harris, a member of the Chapter who celebrated his 76th birthday on the occasion of this meeting.

Western New York

October 10, 1932. This meeting marked the opening of the fall activities of the Western New York Chapter and was held at Gandy's with both members and guests attending.

Pres. M. C. Beman presided and welcomed R. N. Trane of the Wisconsin Chapter, who was present as a guest.

Roswell Farnham appealed to all members in arrears to assist the National Society by paying their dues at this time.

The chairman of the Program Committee, Joseph Davis, promised many interesting meetings during the coming season.

The speaker of the evening was Prof. Grant L. Hector, Ph.D., of the Physics Department of the University of Buffalo, who chose as his subject, Electronic Devices in Industry. Professor Hector described the fundamentals of the vacuum tube and the photo cell and the part played by them in various apparatus in use today. The seemingly magical performances of these devices were very clearly explained by the speaker, who had a thorough understanding of his subject. An interesting phase of the meeting was the lively discussion carried on principally by Professor Hector and L. A. Harding.

The Program Committee is to be commended on securing such an able speaker as Professor Hector.

Pittsburgh

October 17, 1932. The October meeting of the Pittsburgh Chapter was held in the offices of F. C. McIntosh, at 1238 Irwin Ave., N. S., and, instead of the customary hotel dinner, a buffet supper was served by the members at small cost. When Pres. R. B. Stanger called the meeting to order, there were 37 members and guests present.

The minutes of the meeting held May 23 were read and approved.

President Stanger stated that as the question of a reduction in National Society dues was already under consideration by the Council, the dues cut proposal of the Kansas City Chapter, which had been tabled at the May meeting, need not be further considered.

A balance of $200.52 in the treasury was reported by H. B. Orr, treasurer.

F. C. McIntosh, chairman of the Program Committee, said that he expected to have W. S. Brotzman, head of the Pittsburgh Weather Bureau, as the speaker at the November chapter meeting. Mr. McIntosh asked the members what type of meetings they preferred and after discussion it was decided to let the officers arrange for meetings.

President Stanger appointed the following 1933 Nominating Committee, which was requested to present nominations for chapter officers at the November meeting: H. Lee Moore, Chairman, F. H. Hecht and F. C. Houghten.

The speaker of the evening, A. J. Nesbitt, Secy. and Treas. of John J. Nesbitt, Inc., Philadelphia, Pa., was then presented by President Stanger. Mr. Nesbitt gave an illustrated talk on tests he had conducted in a Philadelphia schoolroom in collaboration with J. D. Cassell, Superintendent of Buildings of the Philadelphia Board of Education, describing the principles of good heating and ventilation of schoolrooms and showing methods and results of his experimentation.

After a very interesting discussion, Mr. Nesbitt was given a rising vote of thanks in appreciation of his talk.

There being no further business, the meeting was adjourned at 9:00 p. m.

Illinois

October 10, 1932. A joint meeting of the Illinois Chapter of the A.S.H.V.E. and the Committee of Ten—Coal and Heating Industries—was attended by more than 200 heating and ventilating engineers, coal, coke and equipment men at the Hotel Sherman, Chicago, Ill. The program, which discussed recent advances toward convenience and efficiency in the utilization of solid fuels in domestic heating plants, was arranged and conducted by H. R. Linn and H. M. Hart, both Past Presidents of the Illinois Chapter and also members of the Committee of Ten.

The principal speakers were C. M. Terry, representing the Anthracite Institute, who spoke on Consumer Service; John Jenkins, of the Interlake Iron Corp., who talked on Coke as a Domestic Fuel; and B. R. Gebhart, director of public relations for the Illinois Coal Bureau, who chose for his subject. The New Science of Coal Use.

Mr. Terry left the full development of his subject to Mr. Gebhart and confined his discussion largely to the research and educational work of the Anthracite Institute, which has formed the basis for many of the consumer service activities now engaged in by various solid fuel industries. Particular stress was laid on the development of proper equipment, correct installation and operation, and the important place of the heating engineer in such a field.

Of particular interest to all present was the prediction of Mr. Jenkins that coke manufacturers would profit by experience and give special attention to developing and increasing the use of coke in domestic heating plants. He pointed out that in the past coke manufacturers had used the domestic market to care only for occasional surplus.

Mr. Gebhart seconded Mr. Terry's plea for improved equipment for solid fuel utilization and designated the mechanical stoker as the first real step toward meeting the appeal of automatic heat, which has been fostered by competitive fuels. He briefly reviewed typical instances selected at random from about 50 communities, where complete heat service with solid fuels was being afforded the domestic user either by the retail fuel merchants or through a combination of the fuel merchants and the distributors of stokers and related equipment.

A general discussion followed the talks and among those participating were representatives of various organizations as follows: M. E. Robinson, Jr., *National Retail Coal Merchants Assn.;* R. H. Sherwood, Indianapolis, *Indiana Coal Trade Assn.;* Jack Stowell, *National Warm Air Heating Assn.* and the *National Sheet Metal Contractors' Assn.;* H. H. Kurtz, *Midwest Stoker Assn.;* O. J. Grimes, Committee of Ten; and a number of members of the Illinois Chapter of the A. S. H. V. E.

At the close of the discussion, the annual election of officers of the Illinois Chapter was held and the following newly elected officers were inducted into office:

President—J. H. O'Brien.
Vice-President—C. W. DeLand.
Secretary—J. J. Hayes.
Treasurer—J. H. Milliken.
Board of Governors—R. E. Hattis, G. H. Blanding and J. J. Aeberly.

A. S. M. E. Code for Centrifugal Fans

The *American Society of Mechanical Engineers* Power Test Codes Individual Committee No. 10 on Centrifugal and Turbo-Compressors and Blowers has completed the preliminary draft of the Test Code for Centrifugal and Turbo-Compressors and Blowers.

This Code was one of the group of ten codes forming the 1915 edition of the *A. S. M. E.* Power Test Codes. The code in its present form consists of a complete revision of the 1915 edition.

The personnel of the committee is as follows: A. T. Brown, *chairman,* W. H. Carrier, L. E. Day, E. S. Dean, Z. G. Deutsch, P. E. Good, H. F. Hagen, Arvid Peterson and M. C. Stuart.